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ASHAE Committee on Safety Regulations for Heating, Ventilating and Air Conditioning Systems: G. P. Nachman, *Chairman*; F. H. Buzzard, H. W. McKenzie, B. F. McLouth, C. H. Randolph.

Committee for Revision of ASHAE Standard Code for Testing and Rating Return Line Low Vacuum Heating Pumps: W. M. Wallace, II, *Chairman*; H. E. Adams, H. S. Beam, A. H. Church, L. M. Nechine, E. G. Powell, A. O. Roche, Jr., D. W. Whittington.

* Member of Committee on Research.

General Committees—1957 (*Continued*)

Committee for Revision of ASHAE Standard Code for Testing and Rating Steam Unit Ventilators: John W. McElgin, *Chairman*; L. A. Cherry, A. F. Hubbard, E. F. Hyde, D. H. Krans, L. G. Miller.

Joint ASRE-ASHAE Committee on Testing and Rating Equipment with Respect to Noise: F. O. Urban, *Chairman*; (ASRE) J. B. Chaddock, S. P. Soling; (ASHAE) C. M. Ashley, A. F. Hubbard, R. D. Madison.

Joint Committee on Standards for Comfort Air Conditioning: W. L. Fleisher, *Chairman*; (ASHAE) O. W. Armspach, A. C. Buensod, C. C. Cheyney; (ASME) C. F. Kayan; (ASRE) F. H. Faust, C. F. Holske, S. P. Soling; (ARI) J. A. Gilbreath, R. A. Gonzalez; *Alternate*: G. S. Jones; (*U. S. Public Health Service*) W. C. Cooper, M.D.; *At Large*: M. K. Fahnestock, P. J. Marschall, G. E. May, A. B. Newton; (*Medical Profession*) W. J. McConnell, M.D.

Minimum Ventilation Requirements for Air Conditioned Spaces: Albert Giannini, *Chairman*; C. B. Gamble, R. A. Gonzalez, W. J. McConnell, M.D., R. J. Salinger, H. G. Strong.

Special Committees—1957

AAAS Representative: Walter A. Grant.

ASHAE-AIA Committee on Cooperation: John E. Haines, *Chairman*; C. B. Gamble, John Everetts, Jr., H. C. Hoffmann, R. J. Salinger.

ASHAE-ASRE Committee on Cooperation: ASHAE Representatives—C. B. Gamble, L. N. Hunter, E. R. Queer. **ASRE** Representatives—Cecil Boling, A. J. Hess, H. F. Spoehrer.

Building: A. J. Hess, *Chairman*; M. F. Blankin, L. N. Livermore, P. J. Marschall, A. J. Offner, R. A. Sherman.

Engineers' Joint Council: P. B. Gordon and E. R. Queer (ASHAE Representatives).

Exposition: E. N. McDonnell, *Chairman*; John E. Haines, L. N. Hunter.

Guide Advisory: C. E. Price, *Chairman*; R. E. Cherne, J. C. Fitts, Albert Giannini, J. H. Ross.

Honors and Awards: John E. Haines, *Chairman*; R. A. Sherman, John W. James, H. A. Lockhart, *Ex Officio*.

Gustus L. Larson Award: L. C. Plaehn, *Chairman*; P. D. Gayman, *Co-Chairman*; B. M. Kluge, *Secretary*; Bert Fredericksen, F. W. Goldsmith, F. J. Nunlist, Jr., I. J. Rossiter.

Long-Range Planning: John E. Haines, *Chairman*; E. L. Crosby (*one year*); John Everetts, Jr., L. N. Hunter (*two years*); John W. James, J. Donald Kroeker (*three years*).

NRC—National Academy of Sciences, Div. of Engrg.: John W. James.

Nuclear Energy Engineering: R. A. Sherman, *Chairman*; W. F. Friend, Walter A. Grant, L. N. Hunter, H. A. Lockhart (Committee on Research).

U. S. National Committee of International Institute of Refrigeration: A. J. Hess, ASHAE Representative.

Nominating Committee

Region	*Member*	*First Alternate*	*Second Alternate*
1	C. W. Stone*	W. C. Kruse, Jr.*	E. K. Wagner*
2	H. G. Gragg*	John Richmond*	D. S. Falk*
3	J. S. Locke*	Fred Janssen*	D. M. Allen*
4	R. M. Stern*	H. B. Nottage*	L. F. Christofferson*
5	J. S. Burke*	G. B. Priester*	F. W. Bull*
6	G. A. Linskie*	G. H. Meffert*	G. T. Donceel*
7	D. L. Angus*	A. C. Martin*	G. C. Davis*
3	B. L. Evans†		
4	J. Donald Kroeker†		
6	Reg. F. Taylor†	G. C. F. Asker† (5)	R. L. Stinard† (1)
7	E. K. Cumming†		

* Selected by Chapters Regional Committees. † Selected by Council.

Local Chapter Officers—1957

Arizona

Organized 1953
Headquarters, Phoenix

President . . . . . E. P. Boothroyd
Vice President . . . . . F. W. Gabbard
Secretary . . . . . G. L. Jackson, Jr.
Treasurer . . . . . L. A. Byron, Jr.
Board of Governors: E. P. Boothroyd, L. A. Byron, Jr., F. W. Gabbard, J. B. Hoaglund, G. L. Jackson, Jr.

Arkansas

Organized 1952
Headquarters, Little Rock

President . . . . . H. H. Himstedt
Vice President . . . . . K. A. Pettit
Secretary . . . . . Allen Bullard
Treasurer . . . . . O. L. McCallister, Jr.
Board of Governors: Fred Tenny, T. W. Turner

Atlanta

Organized 1937
Headquarters, Atlanta, Ga.

President . . . . . J. A. Marshall
Vice President . . . . . J. S. Edgar
Secretary . . . . . W. P. West
Treasurer . . . . . E. W. Klein, Jr.
Board of Governors: T. A. Barrow, Jr., F. W. Bull

Austin

Organized 1956
Headquarters, Austin, Texas

President . . . . . A. C. Lagow
Vice President . . . . . J. L. Rea, Jr.
Secretary . . . . . J. R. Watt
Treasurer . . . . . W. H. Brydson
Board of Governors: J. M. Boyer, Fritz Strandtmann

Baltimore

Organized 1949
Headquarters, Baltimore, Md.

President . . . . . R. E. Dressell
Vice President . . . . . F. M. Hewitt
Secretary . . . . . J. H. Baker
Treasurer . . . . . J. F. Malone
Board of Governors: P. L. Harris, W. G. Robertson, Jr., E. K. Schultz, Jr.

Baton Rouge

Organized 1955
Headquarters, Baton Rouge, La.

President . . . . . A. W. Holland
Vice President . . . . . W. J. LeBlanc
Secretary . . . . . J. P. Jones
Treasurer . . . . . R. F. Dupuy
Board of Governors: F. D. Beaullieu, III, A. J. Mayers, Jr.

Bluegrass

Organized 1954
Headquarters, Louisville, Ky.

President . . . . . O. J. Hill
Vice President . . . . . A. E. Handmaker
Secretary . . . . . H. L. Carr
Treasurer . . . . . R. F. Logsdon
Board of Governors: H. J. Bohannon, Jr., S. S. Emison, J. W. Frazier

British Columbia

Organized 1952
Headquarters, Vancouver, B. C., Canada

President . . . . . A. C. Martin
Vice President . . . . . S. W. Welsh
Secretary . . . . . W. F. Wiggins
Treasurer . . . . . D. B. Leaney
Board of Governors: R. E. Atkey, R. W. Hole, D. W. Thomson

Central New York

Organized 1944
Headquarters, Syracuse

President . . . . . Merle Weninger
Vice President . . . . . E. L. Moyer
Secretary . . . . . S. F. Gilman
Treasurer . . . . . E. F. Smarzo
Board of Governors: R. A. Barr, J. W. Stevens

Central Ohio

Organized 1944
Headquarters, Columbus

President . . . . . T. R Walker
Vice President . . . . . R C. Liebert
Secretary . . . . . J. W. Ford
Treasurer . . . . . C. R. Graf**—E. T. Stluka†
Board of Governors: R. A. McGovern, W. A. Schoonover, E. T. Stluka

Chapitre de la Ville de Quebec

Organized 1956
Headquarters, Quebec, P.Q. Canada

President . . . . . Jean Veilleux
Vice President . . . . . P. J. Lamarche
Secretary . . . . . R. A. Harnois
Treasurer . . . . . Maurice Paquet
Board of Governors: L. P. Bonneau, O. Dorval, E. J. Fournier, Maurice Royer, Y. R. Tasse

Cincinnati

Organized 1938
Headquarters, Cincinnati, Ohio

President . . . . . T. D. Reiley
Vice President . . . . . C. P. Krantz
Secretary . . . . . E. A. Sobolewski
Treasurer . . . . . W. H. Rieger
Board of Governors: A. J. Staubitz

** Elected in May 1957
† Elected in September 1957

Local Chapter Officers—1957 (*Continued*)

Connecticut

Organized 1940
Headquarters, New Haven

President . . . . . Fritz Honerkamp
Vice President . . . . . J. D. Pierce
Secretary . . . . . F. J. Raible, Jr.
Treasurer . . . . . R. B. Cahoon
Board of Governors: Walter Heywood, E. M. Johnson, Eric Olsson

Delta

Organized 1939
Headquarters, New Orleans, La.

President . . . . . L. R. Maxwell
Vice President . . . . . R. K. Goode
Secretary . . . . . J. H. Maloney
Treasurer . . . . . O. F. Schully
Board of Governors: D. J. Levine, E. H. Sanford

El Paso

Organized 1956
Headquarters, El Paso, Tex.

President . . . . . G. H. Jackson
Vice President . . . . . W. A. Byars, Jr.
Secretary . . . . . J. H. Brooks
Treasurer . . . . . J. A. True
Board of Governors: W. S. Cooke, Jr., M. D. Goodwin*, J. L. Huff, J. O. LeBeau, Jr.

Empire State Capital

Organized 1951
Headquarters, Albany, N. Y.

President . . . . . H. F. Kruger
1st Vice President . . . . . E. J. Mahoney
2nd Vice President . . . . . M. E. Waddell
Secretary-Treasurer . . . . . B. E. Mullen
Board of Governors: N. J. Ireland, Leonard Levine, E. C. Zyniecki, Jr.

Fort Worth

Organized 1957
Headquarters, Fort Worth, Tex.

President . . . . . C. Rollins Gardner
Vice President . . . . . T. B. Romine, Jr.
Secretary . . . . . J. K. Mattox, Jr.
Treasurer . . . . . C. L. Zahn
Board of Governors: H. E. Palmros, R. M. Williamson

Golden Gate

Organized 1937
Headquarters, San Francisco, Calif.

President . . . . . Herb Duncan
Vice President . . . . . J. B. Smith
Secretary . . . . . T. R. Simonson
Treasurer . . . . . R. C. Pribuss
Board of Governors: F. K. Crouch, D. A. Delaney, L. E. Dwyer

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* Filled unexpired term.

Illinois

Organized 1906
Headquarters, Chicago

President . . . . . H. E. Anderson
Vice President . . . . . Herbert Kreisman
Secretary . . . . . J. C. Scott
Treasurer . . . . . E. P. Heckel, Jr.
Board of Governors: F. A. Balzer, L. O. Paul, J. H. Stiggleman

Illinois-Iowa Chapter

Organized 1956
Headquarters, Moline, Ill.

President . . . . . W. W. Dodge
Vice President . . . . . D. H. Ninow
Secretary . . . . . E. O. Hull
Treasurer . . . . . R. F. Demange
Board of Governors: L. W. Charlet, R. L. Graham, D. G. Johnson

Indiana

Organized 1943
Headquarters, Indianapolis

President . . . . . G. W. Vogel
Vice President . . . . . W. F. Currise
Secretary . . . . . J. M. Teskoski
Treasurer . . . . . A. B. Keller
Board of Governors: R. I. Drum, W. L. Kercheval, A. O. Roche, Jr.

Inland Empire

Organized 1950
Headquarters, Spokane, Wash.

President . . . . . J. A. Doyle
Vice President . . . . . R. D. Nevers
Secretary . . . . . J. R. Morris
Treasurer . . . . . R. B. Campbell
Board of Governors: J. L. Harvey, R. H. Hendron

Iowa

Organized 1940
Headquarters, Des Moines

President . . . . . W. A. Schworm
Vice President . . . . . R. T. Howard, Jr.
Secretary-Treasurer . . . . . C. B. Campbell
Board of Governors: J. L. Bouse, F. L. Pulley, D. E. Schroeder

Jacksonville, Florida

Organized 1957
Headquarters, Jacksonville

President . . . . . B. J. Saymon
Vice President . . . . . J. R. Spence
Secretary . . . . . G. A. Israel, Jr.
Treasurer . . . . . W. P. Sherman
Board of Governors: G. A Israel, Jr., B. J. Saymon, W. P. Sherman, J. R. Spence

Local Chapter Officers—1957 (*Continued*)

Johnstown

Organized 1957
Headquarters, Johnstown, Pa.

President . . . . . W. L. Ross
Vice President . . . . . K. O. Schlentner
Secretary . . . . . J. K. Thornton
Treasurer . . . . . Sigmond Moroh
Board of Governors: H. F. Lenz, L. F. Tierney

Kansas

Organized 1951
Headquarters, Wichita

President . . . . . T. L. Roberts
Vice President . . . . . R. F. Jamison
Secretary . . . . . G. L. Oswald
Treasurer . . . . . L. L. Baker
Board of Governors: A. B. Newton, R. L. Pennington, Charles Yoe

Kansas City

Organized 1917
Headquarters, Kansas City, Mo.

President . . . . . S. C. McCann
Vice President . . . . . J. C. Fasnacht
Secretary . . . . . L. A. Heaven
Treasurer . . . . . T. I. Harriman
Board of Governors: G. F. Hellmer, P. A. Moreno, W. A. Reichow, M. K. Rush, A. J. Sneller, R. M. Spencer

Long Island

Organized 1957
Headquarters, Garden City, L. I., N. Y.

President . . . . . H. J. Campbell, Jr.
Vice President . . . . . C. M. Alston
Secretary . . . . . J. L. Page
Treasurer . . . . . S. M. Walzer
Board of Governors: W. T. Cleland, W. G. Kane, W. H. Hooper

Manitoba

Organized 1935
Headquarters, Winnipeg, Man., Canada

President . . . . . R. M. Fraser
Vice President . . . . . D. S. Swain
Secretary . . . . . J. J. McKenzie, W. L. Algie*
Treasurer . . . . . K. J. McCartney
Board of Governors: G. T. Christie, W. D. Fraser, Stanley Hayden, N. B. Jorgensen, A. H. Millar, A. K. Piercy

Massachusetts

Organized 1912
Headquarters, Boston

President . . . . . E. L. Blair
Vice President . . . . . R. F. Curry
Secretary . . . . . R. E. Reid
Treasurer . . . . . F. J. Butler
Board of Governors: C. A. Biancucci, F. J. Butler, W. R. Geissenhainer, J. P. Hoar, C. T. Mather, C. D. Potter, L. V. Richards, R. B. Stevens, H. F. White

* Filled unexpired term.

Memphis

Organized 1944
Headquarters, Memphis, Tenn.

President . . . . . R. H. Bolding
Vice President . . . . . G. B. Ramsey, Jr.
Secretary . . . . . H. W. Wade
Treasurer . . . . . H. A. Boudolf—J. G. Coleman*
Board of Governors: R. H. Bolding, H. A. Boudolf—J. G. Coleman*, W. L. Henson, G. B. Ramsey, Jr.

Miami Valley

Organized 1950
Headquarters, Dayton, Ohio

President . . . . . L. P. Brehm, Jr.
Vice President . . . . . R. B. Walcott
Secretary . . . . . N. O. Mitchell
Treasurer . . . . . W. H. Fogarty
Board of Governors: J. J. Bechtol, R. J. Perkins, D. E. Tullis, J. B. Rishel, C. D. Weaver, Jr.

Michigan

Organized 1916
Headquarters, Detroit

President . . . . . P. S. Hosman
Vice President . . . . . K. J. Wagoner
Secretary . . . . . J. B. Olivieri
Treasurer . . . . . C. J. Henstock
Board of Governors: J. G. Black, Jr., R. E. Strand, R. E. Maund

Minnesota

Organized 1918
Headquarters, Minneapolis

President . . . . . Fred Vogt
Vice President . . . . . J. H. Jester
Secretary . . . . . J. W. McNamara
Treasurer . . . . . R. G. Gridley
Board of Governors: J. A. Craig, V. E. Pearson

Mississippi

Organized 1953
Headquarters, Jackson

President . . . . . J. W. Doggett, Jr.
Vice President . . . . . T. D. Luke
Secretary . . . . . D. F. Ingram
Treasurer . . . . . W. H. Lambeth
Board of Governors: F. H. North, B. L. Palmer, A. H. Walters

Montreal

Organized 1936
Headquarters, Montreal, Que., Canada

President . . . . . D. J. McIntyre
Vice President . . . . . Arthur de Breyne
Secretary . . . . . Paul Thibault
Treasurer . . . . . W. U. Robinson
Board of Governors: B. J. Horsburgh, W. E. Jarvis, D. L. Lindsay, G. N. Martin, S. R. Plamondon

Nebraska

Organized 1940
Headquarters, Omaha

President . . . . . M. F. Stober
Vice President . . . . . E. N. Seiler
Secretary . . . . . F. P. Manchester, Jr.
Treasurer . . . . . W. L. Ryan
Board of Governors: V. L. Crane, L. J. Paulsen, B. R. Peterson

Local Chapter Officers—1957 (*Continued*)

New Mexico

Organized 1954
Headquarters, Albuquerque

President . . . . . J. K. James
Vice President . . . . . F. H. Bridgers
Secretary . . . . . G. F. Ard**—R. P. Leef†
Treasurer . . . . . E. L. Brazier, Jr.
Board of Governors: W. L. Beale, J. R. Dion, E. T. Kryfko

New York

Organized 1911
Headquarters, New York

President . . . . . C. R. Hiers
Vice President . . . . . W. J. Olvany
Secretary . . . . . L. D. Carr
Treasurer . . . . . W. M. Heebner
Board of Governors: S. R. Apt, P. A. Bourquin, J. B. Hewett, J. M. Levy, S. A. Spencer

North Jersey

Organized 1952
Headquarters, Newark, N. J.

President . . . . . Martin Christesen
Vice President . . . . . S. H. Nitzberg
Secretary . . . . . L. G. Huggins
Treasurer . . . . . C. E. Parmelee
Board of Governors: W. C. Kruse, Jr., H. P. Morehouse, R. S. Taylor, C. W. Zimmer

North Texas

Organized 1938
Headquarters, Dallas

President . . . . . H. G. Gregerson
Vice President . . . . . G. H. Meffert
Secretary . . . . . Ross Zumwalt
Treasurer . . . . . J. D. Poythress
Board of Governors: E. J. Hatzenbuehler, O. H. Mehl, J. A. Ray

Northeastern Oklahoma

Organized 1948
Headquarters, Tulsa

President . . . . . H. W. Meinholtz
Vice President . . . . . J. C. Netherton
Secretary . . . . . H. L. Cauvel
Treasurer . . . . . T. E. Krieg
Board of Governors: C. H. Dollmeyer, J. D Ryan, J. N. Watt, R. W. Winget

Northern Alberta

Organized 1956
Headquarters, Edmonton, Alta., Canada

President . . . . . P. M. Butler
Vice President . . . . . E. K. Cumming
Secretary . . . . . G. N. Campbell
Treasurer . . . . . R. A. Williams
Board of Governors: Harry Hole, J. S. Michener, A. N. Nordstrom, R. G. Proudfoot, Kornelius Siemens

Northern Ohio

Organized 1916
Headquarters, Cleveland

President . . . . . R. E. Leising
Vice President . . . . . R. A. Urban
Secretary . . . . . J. F. Koran
Treasurer . . . . . B. A. Schwirtz
Board of Governors: H. R. Canoyer, R. M. Rubin, R. D. Wilson

** Elected April 1957.
† Elected August 1957

Northern Piedmont

Organized 1952
Headquarters, Greensboro, N. C.

President . . . . . C. A. Cofer
Vice President . . . . . C. E. Amick
Secretary . . . . . B. Jeglinski
Treasurer . . . . . W. H. Cooper
Board of Governors: C. C. Donovan, L. R. Gorrell, G. B. Rottman

Oklahoma

Organized 1935
Headquarters, Oklahoma City

President . . . . . A. C. Shelley
Vice President . . . . . W. R. Johnson
Secretary . . . . . W. H. Stewart
Treasurer . . . . . T. L. Robinson
Board of Governors: E. F. Dawson, J. R. Patten

Ontario

Organized 1922
Headquarters, Toronto, Ont., Canada

President . . . . . C. L. Torry
Vice President . . . . . Jack Thompson
Secretary-Treasurer . . . . . H. R. Roth
Board of Governors: C. J. Bootes—D. L. Angus*, J. D. Coates, Ernest Fox, R. A. Ritchie

Oregon

Organized 1939
Headquarters, Portland

President . . . . . W. B. Hayes
Vice President . . . . . Keith Kruchek
Secretary . . . . . W. S. Cooper
Treasurer . . . . . G. H. Lord
Board of Governors: W. R. Norte, T. E. Taylor, E. S. Constant

Ottawa Valley

Organized 1952
Headquarters, Ottawa, Ont., Canada

President . . . . . C. W. Watson
Vice President . . . . . Jacob Klassen
Secretary . . . . . A. H. Hargreaves
Treasurer . . . . . S. J. Bullis
Board of Governors: J. W. Green, C. N. Kirby, S. C. Smith

Philadelphia

Organized 1916
Headquarters, Philadelphia, Pa.

President . . . . . P. H. Yeomans
1st Vice President . . . . . C. J. Forve
2nd Vice President . . . . . J. J. Hucker
Secretary . . . . . D. S. Plewes
Treasurer . . . . . Ludwig Mack
Board of Governors: John Everetts, Jr.

Pittsburgh

Organized 1919
Headquarters, Pittsburgh, Pa.

President . . . . . E. H. Riesmeyer, Jr.
Vice President . . . . . E. J. Busch
Secretary . . . . . G. E. Smetak
Treasurer . . . . . R. C. Firsching
Board of Governors: G. W. Cost, A. F. Nass, Jr., R. M. Toucey

* Filled unexpired term.

Local Chapter Officers—1957 (*Continued*)

Puget Sound

Organized 1928
Headquarters, Seattle, Wash.

President.....E. W. Triol
1st Vice President.....M. W. McKinstry
2nd Vice President.....H. M. Hendrickson
Secretary.....R. H. Liniger
Treasurer.....K. G. Massart
Board of Governors: E. T. Franklin, Jr., D. M. Hopkins

Rocky Mountain

Organized 1944
Headquarters, Denver, Colo.

President.....D. D. Pearsall
Vice President.....J. R. Wilson
Secretary.....J. F. Perkis
Treasurer.....M. D. Beckett
Board of Governors: J. E. Harrington, F. E. Stark, A. S. Widdowfield

Sacramento Valley

Organized 1952
Headquarters, Sacramento, Calif.

President.....M. J. Delavan
Vice President.....J. A. White
Secretary.....R. T. Andrews
Treasurer.....G. F. Romack
Board of Governors: W. B. Lander, R. J. McKinnon—E. J. Varvello*, V. W. Thornburg

St. Louis

Organized 1918
Headquarters, St. Louis, Mo.

President.....N. J. Hubbuch
Vice President.....J. J. Blackmore
Secretary.....K. O. Williams
Treasurer.....J. B. Killebrew
Board of Governors: C. W. Baker, E. C. Kuntz, J. I. Levenhagen, W. P. Norris, G. B. Pattiz, R. R. Waites

Shreveport

Organized 1948
Headquarters, Shreveport, La.

President.....A. L. Jones, Jr.
Vice President.....L. E. Kneipp
Secretary.....J. L. Collins, Jr.
Treasurer.....J. J. Guth, Jr.
Board of Governors: H. E. Scott, T. F. Scott, R. F. Zimmerman

South Carolina

Organized 1954
Headquarters, Columbia

President.....B. A. Leppard
Vice President.....A. L. Blackstone, Jr.
Secretary.....H. J. Haar, Jr.
Treasurer.....A. M. Suggs
Board of Governors: F. A. Bailey, III, J. E. McMurray, W. O. Blackstone

South Texas

Organized 1938
Headquarters, Houston

President.....J. C. Lewis
Vice President.....A. B. Ullrich, Jr.
Secretary.....E. B. Appling
Treasurer.....W. A. Sharp
Board of Governors: H. D. McMillan, Jr., J. H. Stephenson

* Filled unexpired term.

Southern California

Organized 1930
Headquarters, Los Angeles

President.....W. J. Biggar
Vice President.....A. Z. Levine
Secretary.....R. W. Thomas—R. C. Taylor*
Treasurer.....W. L. Holladay
Board of Governors: G. D. Lord, H. B. Nottage, R. C. Taylor—K. D. Simon, H. F. Ulovec, Albert Zimmerman

Southern Piedmont

Organized 1952
Headquarters, Charlotte, N. C.

President.....G. C. Garrett
Vice President.....J. R. Clark
Secretary.....N. W. McGuire, Jr.
Treasurer.....E. D. Streng
Board of Governors: A. W. Bainbridge, G. N. Payne, Jr., J. A. Rice

Southwest Texas

Organized 1946
Headquarters, San Antonio

President.....C. L. Herndon
Vice President.....D. T. Kern
Secretary.....L. A. Ingle, Jr.
Treasurer.....Martin Royer
Board of Governors: E. E. Cravens, Boone Crisp, W. F. Markey

Toledo

Organized 1954
Headquarters, Toledo, Ohio

President.....Robert Greenwald
Vice President.....D. L. Wilson
Secretary.....G. R. Munger
Treasurer.....J. E. Wilkie
Board of Governors: J. P. Eyster, J. F. Guest, E. J. Katafiasz, M. D. Williams

Utah

Organized 1944
Headquarters, Salt Lake City

President.....Alfred Richeda
Vice President.....L. K. Irvine
Secretary-Treasurer.....E. V. Gritton
Board of Governors: A. R. Curtis, R. C. Evans, W. L. Stuewe

Virginia

Organized 1946
Headquarters, Norfolk

President.....D. C. Delinger
Vice President.....D. L. Gusler
Secretary.....J. A. Hoffman
Treasurer.....W. C. Andrews
Board of Governors: J. F. Boyenton, A. R. Thompson, Jr.

Washington, D. C.

Organized 1935
Headquarters, Washington, D. C.

President.....H. W. Rush
Vice President.....J. W. Morgan
Secretary.....L. D. Pain
Treasurer.....R. F. Dovener
Board of Governors: J. H. Broome, V. L. Frederick, H. E. Grossman

* Filled unexpired term

Local Chapter Officers—1957 (*Continued*)

West Texas

Organized 1953
Headquarters, Lubbock

President.....H. L. Mayes
Vice President.....V. B. Garry
Secretary.....W. R. Anthony
Treasurer.....C. P. Houston
Board of Governors: R. L. Mason, J. C. Wharton

Western Massachusetts

Organized 1955
Headquarters, Springfield

President.....F. A. Ferraro
Vice President.....C. R. Munn
Secretary.....J. J. Curran
Treasurer.....K. W. Maki
Board of Governors: R. E. Cross, J. F. Hayes, A. M. Lovenberg, J. E. Reed, B. R. Washburn

Western Michigan

Organized 1931
Headquarters, Grand Rapids

President.....B. J. Walter
Vice President.....R. J. Waalkes
Secretary.....S. R. Curtis
Treasurer.....F. K. Platt
Board of Governors: W. C. Harrower, P. S. Morton, F. L. Murray

Western New York

Organized 1919
Headquarters, Buffalo

President.....G. E. Kuhn
1st Vice President.....J. P. Guerra
2nd Vice President.....J. F. Bedard
Secretary.....D. J. Seifert
Treasurer.....M. C. Beman
Board of Governors: Joseph Davis, Roswell Farnham, V. N. Harwood, C. W. Stone, Q. P. Thompson

Wisconsin

Organized 1922
Headquarters, Milwaukee

President.....H. W. Alyea
Vice President.....R. D. Rodwell
Secretary.....A. A. Stutheit
Treasurer.....H. W. Snyder
Board of Governors: H. K. Forfar, B. M. Kluge, C. E. Melcher

Overseas Branch

Switzerland

Organized 1952
Headquarters, Zurich

President.....Albert Eigenmann
Vice President.....R. A. Goerg
Secretary.....Hans Kamm
Treasurer.....Werner Niederer
Controller.....Werner Gysi

Student Branches

North Carolina State College

Organized 1948
Headquarters, Raleigh

Faculty Adviser.....Prof. R. B. Knight

Texas A & M College

Organized 1946
Headquarters, College Station

Faculty Adviser.....Prof. L. S. O'Bannon

Oregon State College

Organized 1949
Headquarters, Corvallis

Faculty Adviser.....Prof. G. E. Thornburgh

University of Detroit

Organized 1949
Headquarters, Detroit, Mich.

Faculty Adviser.....J. B. Olivieri

Purdue University

Organized 1949
Headquarters, W. Lafayette, Ind.

Faculty Adviser.....Prof. F. B. Morse

University of Toronto

Organized 1951
Headquarters, Toronto, Ont., Canada

Faculty Adviser.....Prof. F. G. Ewens

TRANSACTIONS

of

AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS

No. 1590

SIXTY-THIRD ANNUAL MEETING, 1957

CHICAGO, ILLINOIS

Attendance records were broken and new features appeared on the program of the 63rd Annual Meeting held at Chicago, February 25–28, 1957 with headquarters at the Conrad Hilton Hotel. There were 6 technical sessions with 2 taking the form of symposiums, and concurrent sessions were held in order to provide sufficient time to accommodate the sessions.

On the attendance side, the total registration was 3257, made up of 1808 members, 954 guests and 495 ladies. The 13th International Heating and Air Conditioning Exposition held at the International Amphitheater during the same week as the Meeting not only attracted 472 exhibitors but also an estimated attendance in excess of 30,000.

In addition to the 6 technical sessions with 13 papers, the program of the Meeting (see page 22) included numerous meetings of Technical Advisory Committees and of general and special committees of the Society. There was a welcome luncheon with Dr. Richard Heflebower, Northwestern University, as speaker; a talent and tea for the ladies; an evening Get-Together Party at the hotel; inspection trips; a country-club luncheon for ladies; and the annual banquet at which Dr. Preston A. Bradley, Pastor of People's Church, Chicago, was the featured speaker. The Past Presidents held their annual dinner, and life members for the second year met at a special luncheon.

Honors and awards were featured with the first award of the ASHAE-Homer Addams Award being presented to Paul M. Chung, a graduate student at the University of Minnesota. The fifteenth award of the F. Paul Anderson Medal went to Professor M. K. Fahnestock, University of Illinois. Five members received the grade of Fellow, granted for the first time. Past Pres. John E. Haines presented the Past Presidents Award to retiring Pres. John W. James. The meeting was also notable because of the presence of Past Pres. Arthur C. Willard, President Emeritus, University of Illinois who was elected as an Honorary Member.

FIRST SESSION, MONDAY, FEBRUARY 25, 9:30 A.M.

Meeting in the Waldorf Room of the Conrad Hilton Hotel, Chicago Ill., Pres. John W. James, Chicago, Ill., called the session to order. He at once presented P. J. Marschall, North Chicago, Ill., general chairman of the Committee on Arrangements, who expressed the pleasure of the Chicago Chapter in welcoming the meeting. Mr. Marschall also outlined the program features as arranged for the meeting and asked that all attend as many of them as possible.

REPORT OF PRESIDENT

President James then read his report in which he mentioned many of the outstanding events which had taken place during the year. He announced a steady growth in membership, that the Society activities in research and standards were progressing, that the public relations program was organized and advancing, and that generally the affairs of the Society were on a firm basis. (The full text of the Report of the President can be found on pages 143 to 145 of the ASHAE JOURNAL SECTION, *Heating, Piping and Air Conditioning*, May, 1957).

As the next order of business, President James asked that A. V. Hutchinson, New York, N. Y., executive secretary, present his report. Mr. Hutchinson at this time also presented the report of the Council.

REPORT OF THE COUNCIL

The 1956 annual report of the Council of the Society as required by the Membership Corporations' Law, is presented herewith:

Four regular meetings of the Council were held during the calendar year, 1956, the organization meeting being convened January 26 at the Sheraton-Gibson Hotel, Cincinnati, Ohio. Personnel of the Council, General and Special Committees authorized by the By-Laws were announced by Pres. John W. James.

At the organization meeting, the Council approved 7 Regional Areas for chapter operations and specified the chapters in each region. Among the important actions taken during the year by the Council were:

1. Approved Charters for 6 chapters.
2. Recommended honorary membership for 3 persons of professional distinction and nominated a fourth for honorary membership in 1957.
3. Conferred the new grade of Fellow upon 5 members.
4. Elected 15 members to Life Membership.
5. Approved an official definition of air conditioning.
6. Authorized the first ASHAE-Homer Addams Award.
7. Voted that Prof. Maurice K. Fahnestock be awarded the F. Paul Anderson Award for 1956.

8. Approved the Operational Guide for the Gustus L. Larson Award submitted by Wisconsin Chapter, and authorized the first Larson Award.

9. Approved the Operational Guide for the Long Range Planning Committee.

10. Noted the Society's election as a member of the *Engineers Joint Council* and named First Vice-Pres. P. B. Gordon and Second Vice-Pres. E. R. Queer to serve respectively as the Society's representative and alternate on the Board of *EJC*.

New By-Laws on membership grades to conform to *ECPD* recommendations, submitted to the membership in December 1955, as well as proposed changes in chapter operations and Nominating Committee procedure, were adopted.

The membership changes included establishment of the honorary grade of Fellow and the designation of all Junior Members as Associate Members.

Other changes were the outlining of a regional plan for chapter operations; changes in dues to conform to the revised membership grades; providing for 7 Council members to serve as regional directors; providing for a new Council Committee (Regions Central Committee) and a new General Committee (Chapters Regional Committee) to replace, respectively, the Chapter Relations Committee and the Chapters Conference Committee; a minor revision in the procedure of the Admission and Advancement Committee; and a new term of office for the Guide Committee in order to increase its effectiveness and, an increase in the number of alternates on the Nominating Committee and provision for payment of members' transportation to 2 meetings instead of to 1.

Charter Grants During 1956: (1) Northern Alberta Chapter (Canada)—Approved January 22, 1956; (2) Austin Chapter (Tex.)—Approved April 8, 1956; (3) El Paso Chapter (Tex.)—Approved June 17, 1956; (4) Illinois-Iowa Chapter (Ill.)—Approved June 17, 1956; (5) Chapitre de la Ville de Quebec (Canada)—Approved October 21, 1956; (6) Fort Worth Chapter (Tex.)—Approved October 21, 1956.

Honorary Membership: Elected to Honorary Membership were the Honorable Herbert Hoover, New York, N. Y.: Dr. Milton S. Eisenhower, President, Johns Hopkins University, Baltimore, Md.; Dr. Charles F. Kettering, researcher, inventor and industrialist, Dayton, Ohio. Proposed for Honorary Membership in 1957 was Dr. Arthur C. Willard, President Emeritus, University of Illinois, Urbana, Ill.

Life Membership: Members elected to Life Membership were M. C. Beman, H. J. Church, G. M. Comstock, Carl H. Flink, I. H. Geiger, D. C. Jardine, H. L. Janet, R. L. Leilich, N. J. Mathey, C. R. Morris, C. F. Olsen, J. L. Pfuhler, H. P. Reid, R. B. Stanfer, M. S. Wunderlich.

Fellows: The Council on October 21, 1956, conferred the grade of Fellow upon Walter L. Fleisher, Sr., L. N. Hunter, J. Donald Kroeker, Charles S. Leopold and Samuel R. Lewis.

Definition of Air Conditioning: On April 8, 1956 the Council approved an official definition of air conditioning.

ASHAE-Homer Addams Award: The first Homer Addams Award was authorized on June 17, 1956. It was presented during the 1957 Annual Meeting to Paul M. Chung, St. Paul, native Korean and graduate student of the University of Minnesota.

Gustus L. Larson Award: The first Gustus L. Larson Award, approved by Council October 21, 1956, was made to Ronald L. Kausch, senior in mechanical engineering at the University of Wisconsin, on November 15, 1956, at a joint dinner meeting of the Rock River Valley Section and the University of Wisconsin Section of *ASME*.

Deaths: Two past presidents of ASHAE passed away during 1956, and one died during January of this year. They were Dr. B. M. Woods, September 7, 1956, Ernest Szekely, November 29, 1956, and Dr. C.-E. A. Winslow, January 8, 1957.

Finances: As required by Section 46 of the Membership Corporations' Law of the State of New York the following report is presented by the Council and filed herewith:

Assets—October 31, 1956

The Society has the following Assets:

Land and Laboratory Buildings at Cleveland, Ohio	$ 81,012.89
Furniture and Equipment, Tools, etc., at New York and Cleveland	39,054.60
On deposit in New York and Cleveland banks	214,257.50
Securities (United States and Utility Bonds) with Bankers Trust Co., New York as custodian	133,294.48
Accounts Receivable	12,764.89
Inventory Items	8,251.98
Deferred Items	12,254.66
TOTAL ASSETS—1956	$500,891.00*
TOTAL ASSETS—1955	494,929.14
Increase for the year	$ 5,961.86

Liabilities—October 31, 1956

The Society has the following Liabilities:

Accounts Payable	$ 238.17
Federal Withholding Taxes	1,457.04
Accrued Accounts	15,580.25
Deferred Income:	
Research Projects	10,556.16
Prepaid Dues, Initiation Fees, etc.	13,272.17
Reserve for Fluctuation in Canadian Exchange	35.29
TOTAL LIABILITIES	$ 41,139.08
Net Worth (Funds)	459,751.92
TOTAL LIABILITIES AND NET WORTH	$500,891.00

At the end of the fiscal year, October 31, 1956, the Balance Sheet shows the Society's net worth as $459,751.92 compared to $422,190.70 on October 31, 1955, a net gain of $37,561.22.

Total cash in New York and Cleveland banks $214,257.50 consisting of $54,764.28 in General Operating Fund, $38,112.52 in Reserve Fund, $321.44 in F. Paul Anderson Fund, $72,249.42 in Research General Fund, $28,639.62 in Research Reserve, $777.26 in Research Endowment, $5,974.13 in Building Maintenance Reserve Fund, $3,346.49 in Building Fund, $10,022.12 in the ASHAE-Homer Addams Fund, and $50.22 in the Educational Fund, Life Members.

Now in the Custodian Account of the Bankers Trust Co., New York, there are securities of the United States and Canadian Governments and a utility which cost as follows: General Fund $3,782.36; Building Fund $35,462.00; Reserve Fund $88,166.50 and F. Paul Anderson Fund $1,000.00.

The Society's Reserve Fund was increased by the addition of $9,741.00 Admission Fees and $2,452.13 Interest earned on Savings Bank Accounts and Securities. This is an increase of $12,193.13 bringing the Reserve Fund to $130,119.29. The Research Fund was increased by the addition of $1,270.75 Interest Earned on Savings Accounts.

* After elimination of $5,459.97 intra-fund receivable and payable.

During the year $25,000.00 of this fund was transferred to Research General Fund. As of October 31, 1956 the Research Reserve Fund had a balance of $28,639.62.

The Sources of Society Income were: Admission Fees and Dues $158,561.13; Publications $227,625.60; Interest $664.27; From Research $208,398.05; Other Income $1,197.01—Total $596,446.06.

Society Expenditures were: Meetings, Committees and Chapters $31,252.86; Publications $140,489.34; Headquarters $173,484.59; Research $218,187.38—Total $563,414.17.

The insurance coverage of the Society is as follows: Fire for Buildings $201,000.00; for Contents $144,500.00; Sprinkler $12,000.00; Automobile Comprehensive Public Liability $100,000–300,000.00; Property Damage $25,000.00; Medical Payments $1,000.00; Bodily Injury $25,000–100,000.00; Public Liability, Premises, New York and Cleveland $100,000–300,000.00; Fidelity Bonds on Officers and all Employees $20,000.00. Workmen's Compensation is carried in New York and Ohio as required by law.

The Society has contributed to the Employees Retirement Plan Trust $2,875.61 and the participating employees contributed a substantially similar amount deducted from salary.

Membership

The Society admitted the following members in the grades indicated since January 1, 1956:

Life Members (Transfer 15)	
Members	336
Affiliates	318
Associates	456
Junior Members* (January 1956 selection)	22
Students	176
Total	1,308

During the period January 1, 1956 to January 15, 1957, a total of 20 members in the Member, Associate Member, and Affiliate grades were reinstated; a total of 423 in all categories were dropped from the rolls; 184 resigned, and death claimed 56.

Net gain of members was 665. The overall membership status from January 1, 1956 as compared with January 15, 1957 is as follows:

	January 1, 1956	January 15, 1957
Honorary	1	4
Presidential Members	24	21
Life Members	218	226
Fellows	—	5
Members	4,424	4,691
Associate Members	1,813	4,115
Affiliates	1,752	1,876
Junior Members*	2,120	—
Students	124	203
	10,476	11,141

The names and addresses of the candidates for membership were published in the JOURNAL of the Society each month during the year and are on record in the Secretary's office. The present membership total is 11,141.

Respectfully submitted,
JOHN W. JAMES, *President*
RALPH A. SHERMAN, *Treasurer*

* Automatically became Associate Members upon change in By-Laws adopted at 62nd Annual Meeting Cincinnati, Ohio, January 26, 1956.

REPORT OF TREASURER

The treasurer of the Society, R. A. Sherman, Columbus, Ohio, then reported to the meeting. His material consisted of two parts, *viz*: a summary of the operations of the treasurer's office and also the Accountant's Report which appears on pages 6 to 11.

In his report of the treasurer's operations, Mr. Sherman indicated that from a Society gross income of $596,446.06, there was allocated to Research:

Dues (40% of Members, Associates over 30, and Affiliates)	$ 79,491.75	
Research contributions		
Received in fiscal year	115,534.63	
Deferred from prior years	12,890.32	
Interest	481.35	
		$208,398.05

To this, the Council added $25,000 from the Research Reserve, from the 1955 Exposition, making a total available for Research of $233,398.05. Thus, based on the receipts of the current year, the funds available for research were 35 percent of the current total Society receipts; including the allotment from the Research Reserve, they were 37.5 percent of the current receipts.

Accountant's Report

FRANK G. TUSA & CO.

CERTIFIED PUBLIC ACCOUNTANTS

37 Wall St., New York 5, N. Y.

AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC.
62 Worth St., New York, N. Y.

Gentlemen:

In accordance with the authority contained in the minutes of the organizational meeting of the 1956 Council on January 26, 1956, we have examined the books of account of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC.—New York and Cleveland for the fiscal year ended October 31, 1956.

Our examination consisted of the verification of assets and liabilities as of the close of business October 31, 1956 and a review of the operations for the year then ended. The examination was conducted in accordance with generally accepted auditing standards, except as otherwise noted in the comments section and included such aduiting procedures and such other tests of the accounting records as we considered necessary under the circumstances.

We are pleased to present our report of this examination which includes the various statements and schedules listed in the table of contents in addition to the following comments that constitute an integral part of this report:

ASSETS

CASH (Page 13)

On Hand	$ 150.00	
In Checking Accounts	89,864.49	
In Savings Banks	124,243.01	$214,257.50

The cash on hand was physically verified. The cash on deposit was verified by direct communication with all depositories, and the balances reported to us were reconciled to those reflected on the books of account of the Society and found to be correct. A schedule of cash balances in all funds is included on page 13 as a part of this report.

INVESTMENTS AT COST (Page 14)	$128,410.86

The securities shown on the subjoined schedule were verified by direct communication with the Bankers Trust Company, where same are deposited for safe-keeping. Securities have been included in the accompanying Statement of Financial Condition at the cost of acquisition plus the accumulated and accrued interest earned thereon.

ACCOUNTS RECEIVABLE	$ 22,125.21

Trial balances taken of the membership dues receivable and sundry debtors as of the close of business October 31, 1956 were classified and aged as follows:

MEMBERSHIP

Members	$ 5,357.79
Associate Members (over 30 years)	3,490.25
Affiliates	4,049.00
Associate Members (under 30 years)	2,064.75
Students	15.00
TOTAL	$ 14,976.79

SUNDRY DEBTORS

Unpaid Charges made during 1956	$ 7,148.42

No outside confirmation was requested of accounts receivable. We were, however, by means of other auditing procedures, able to confirm, substantively, the above balances which in our opinion are correct. Consideration was given to all possible uncollectible accounts, and it is our opinion that present reserves for such uncollectibles are ample to cover collection losses that may be incurred.

RESEARCH FUND ACCOUNTS RECEIVABLE

The balance due to the Research Fund from the Society as at October 31, 1956 represents the unpaid balance of 40 percent of the dues receivable from members, associates and affiliates as of such date.

INVENTORIES

The following inventories were verified either by physical count or by direct communication with the appropriate printers.

TRANSACTIONS	$ 7,064.95
EMBLEMS	1,079.10
GUIDE Paper	107.93
TOTAL	$ 8,251.98

A Schedule of TRANSACTIONS inventories follows:

Volume	Year	Quantity	Price	Amount
1–55	1895–1949	4,757	.4000	$1,902.80
56	1950	670	2.0800	1,393.60
57	1951	656	2.2750	1,492.40
58	1952	140	1.7960	251.44
59	1953	415	2.0000	830.00
60	1954	273	2.0665	564.15
61	1955	304	2.0742	630.56
TOTAL				$7,064.95

DEPOSITS RECEIVABLE	$425.00

The deposit placed with the United Airlines in the sum of $425.00 was verified by direct communication.

ADVANCES	$9,060.44

The indebtedness from employees in New York and Cleveland represents advances to the employees retirement plan in the sums of $7,720.67 and $1,339.77 respectively.

PREPAID EXPENSES AND DEFERRED CHARGES	$ 2,769.22

The following prepayments have been deferred to future operations:

H.P.A.C.—Subscriptions	$ 619.16
Insurance Premiums—Cleveland	1,967.96
Insurance Premiums—New York	182.10
TOTAL	$ 2,769.22

PERMANENT ASSETS (Page 12)	$120,067.49

The Land and Buildings, Instruments, Equipment and Furniture and Fixtures are reflected on the Statement of Financial Condition at cost of acquisition. With the exception of land and buildings all assets have been depreciated at the rate of 10% per annum.
In accordance with the resolution adopted by the council at its meeting of January 23, 1949 depreciation was not provided on the buildings for the current fiscal year.

DEFERRED INCOME	$ 10,232.17

The prepaid dues and initiation fees by members and candidates for membership have been deferred to future operations. The membership classification of the dues prepaid by elected members follow:

MEMBERSHIP

Members	$ 5,917.51
Associate Members (over 30 years)	2,281.25
Affiliates	1,185.33
Associate Members (under 30 years)	796.08
Students	52.00
TOTAL	$ 10,232.17

RESERVE FOR BUILDING MAINTENANCE	$ 5,974.13

On October 31, 1952, the Council voted:

"That disbursements be made from the Building Maintenance Reserve Fund as authorized by the Building Committee for major items of repair and maintenance, but any balance remaining in the fund beyond the expenditures for each year would go into reserve until the Building Maintenance Reserve Fund reached a total of $10,000.00."

The Building Maintenance Reserve Fund had a balance at the beginning of the current fiscal year of $5,844.99. Appropriate additions and deductions have been made within the fund in accordance with budgetary provisions of the current fiscal year.

FUNDS	$459,751.92

There is included as a part of this report a Statement of Funds reflecting the changes that occurred therein during the fiscal year ended October 31, 1956.

INSURANCE

The insurance coverage of the Society follows:

FIRE		
Building (Cleveland)		$201,000.00
Personal Property:		
Cleveland	$104,000.00	
Printers	15,000.00	
New York	25,500.00	144,500.00
SPRINKLER LEAKAGE		
Waverly Press	10,000.00	
Hawes and Petit	2,000.00	12,000.00
NON-OWNERSHIP—AUTOMOBILE—COMPREHENSIVE		
Public Liability		100/300M
Property Damage		25M
Medical		1M
GENERAL PUBLIC LIABILITY—PREMISES		
New York and Cleveland		
Liability		100/300M
Bodily Injury		25/100M
COMMERCIAL BLANKET BOND		
All Employees including the President, Treasurer and Chairman of the Finance Committee		20,000.00

In our opinion, the accompanying Statement of Financial Condition and Statement of Income and Expenses present fairly the financial condition of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC. at October 31, 1956, and the results of its operations for the year then ended, in conformity with generally accepted accounting principles applied on a basis consistent with that of the preceding year.

Dated December 1, 1956 FRANK G. TUSA & CO., CERTIFIED PUBLIC ACCOUNTANTS

STATEMENT OF FINANCIAL CONDITION—October 31, 1956

ASSETS

GENERAL FUND			
Society Assets (Per Schedule)			$ 88,507.88
Research Assets (Per Schedule)			81,047.18
PROPERTY FUND (Per Schedule)			120,067.49
BUILDING FUND			
Cash on Deposit		$ 3,346.49	
Investments			
Securities at Cost	$ 35,462.00		
Plus Accumulated Interest	1,003.35	36,465.35	39,811.84
BUILDING MAINTENANCE RESERVE FUND			
Cash on Deposit			5,974.13
SOCIETY RESERVE FUND			
Cash on Deposit		38,112.52	

Investments			
Securities at Cost	88,166.50		
Plus Accumulated Interest	3,840.27	92,006.77	130,119.29
EDUCATIONAL FUND (LIFE MEMBERS)			
Cash on Deposit			50.22
ENDOWMENT FUNDS			
F. Paul Anderson Award Fund			
Cash on Deposit		321.44	
Investments			
Securities at Cost	1,000.00		
Plus Accumulated Interest	12.50	1,012.50	1,333.94
Research Fund			
Cash on Deposit			777.26
Homer Addams Fund			
Cash on Deposit			10,022.12
RESEARCH RESERVE FUND			
Cash on Deposit			28,639.62
TOTAL ASSETS			$506,350.97

LIABILITIES

GENERAL FUND			
Federal Withholding Taxes		1,457.04	
Due To Research General Fund		5,459.97	
Accrued Accounts			
Salaries and Commissions		14,330.25	
Professional Fees		1,250.00	
Sundry Account Payable		100.00	
Deferred Income			
Prepaid Membership Dues			
Elected Members	$ 6,478.54		
Candidates	831.00	7,309.54	
Prepaid Admissions Fees		2,209.00	
Allowance For Fluctuation in Canadian Exchange		35.29	$ 32,151.09
RESEARCH FUND			
Sundry Accounts Payable		138.17	
Deferred Income			
Unexpended Ear-Marked Contributions	6,556.16		
Prepaid Dues	3,753.63		
Special Contribution	4,000.00	14,309.79	$ 14,447.96

NET WORTH

Society General Fund	56,356.79	
Research General Fund	66,599.22	
Property Fund	120,067.49	
Building Fund	39,811.84	
Building Maintenance Reserve Fund	5,974.13	
Society Reserve Fund	130,119.29	
F. Paul Anderson Award Fund	1,333.94	
Research Endowment Fund	777.26	
Research Reserve Fund	28,639.62	
Homer Addams Fund	10,022.12	
Educational Fund (Life Members)	50.22	459,751.92
TOTAL LIABILITIES AND NET WORTH		$506,350.97

CONDENSED COMPARATIVE STATEMENT OF INCOME AND EXPENSES

	OCT. 31, 1956	OCT. 31, 1955	INCREASE (DECREASE)
INCOME			
DUES	$148,820.13	$140,413.30	$ 8,406.83
ADMISSION FEES	9,741.00	8,712.65	1,028.35
EMBLEMS, PINS, ETC.	961.80	1,558.75	(596.95)
PUBLICATIONS			
JOURNAL CONTRACT	28,916.70	23,500.00	5,416.70
THE GUIDE	187,744.02	163,552.46	24,191.56
TRANSACTIONS	1,704.20	1,358.08	346.12
Books, Reprints, Codes, etc.	5,314.72	4,431.74	882.98
Membership Roll	3,945.96	7,685.50	(3,739.54)
INVESTMENTS	664.27	908.55	(244.28)
FOREIGN EXCHANGE	65.31	166.79	(101.48)
NON-RECURRING	169.90	-0-	169.90
RESEARCH			
Dues	79,491.75	73,967.74	5,524.01
U. S. Navy Research	127.50	1,599.68	(1,472.18)
Contributions—General	22,079.63	24,319.71	(2,240.08)
Contributions—Earmarked	106,217.82	82,934.45	23,283.37
Interest	481.35	502.78	(21.43)
TOTAL INCOME	596,446.06	535,612.18	60,833.88
EXPENSES			
COMMITTEES AND CHAPTERS	24,595.87	29,869.95	(5,274.08)
MEETINGS	6,656.99	6,884.10	(227.11)
PUBLICATIONS			
Member Subscriptions to *HPAC*	3,771.25	20,042.06	(16,270.81)
TRANSACTIONS			
Volume 61	24,889.85	-0-	24,889.85
Volume 60	2,895.96	26,004.03	(23,108.07)
Membership Roll	2,228.24	5,796.67	(3,568.43)
Books, Reprints, Codes, etc.	3,936.29	2,604.57	1,331.72
THE GUIDE	102,767.75	98,031.01	4,736.74
HEADQUARTERS	173,484.59	170,013.83	3,470.76
FUND RAISING	19,375.65	16,936.79	2,438.86
RESEARCH			
Committee Expenses	3,435.14	5,168.06	(1,732.92)
Staff Salaries	107,862.52	100,175.04	7,687.48
Laboratory Expenses	25,598.10	23,791.87	1,806.23
Building Operation and Maintenance	11,915.97	9,804.23	2,111.74
Provision for Building Maintenance Reserve	2,000.00	2,000.00	-0-
U. S. Navy Research	-0-	676.51	(676.51)
Cooperative Research	48,000.00	44,821.70	3,178.30
OTHER DEDUCTIONS	-0-	379.64	(379.64)
TOTAL EXPENSES	563,414.17	563,000.06	414.11

President James then called attention to the point that Dr. Arthur Cutts Willard, past president of the Society, had been nominated to be an Honorary Member. This nomination was made at the fall meeting of the Council and the By-Laws provide for the election of such a nominee at the Annual Meeting. President James called for a voice vote and Doctor Willard was elected an Honorary Member.

President James expressed his pleasure that J. Clifford Byles, president of the *Institution of Heating and Ventilating Engineers*, London, England, was present and asked him for a few words of greeting. Mr. Byles responded with greetings and good wishes from the *Institution* and the members and Council.

President James also called on F. R. L. White, a past president of the *Institution*, and a Member of *ASHAE*. Mr. White responded with remarks concerning the English method of training students in a national college.

Three technical papers had been programmed for this session and President James called on their authors for the presentation. Following presentation of the first two papers, President James announced the appointment of the Inspectors of Election and the Committee on Resolutions, stating that both groups would report at later sessions. After the completion of the reading of the third technical paper, President James closed the session at 12:00 noon.

SECOND SESSION, TUESDAY, FEBRUARY 26, 9:30 A.M.

As soon as he had called the session to order in the Waldorf Room at the Conrad Hilton Hotel, 1st Vice Pres. P. B. Gordon, New York, N. Y., announced that Prof. B. H. Jennings, Evanston, Ill., and chairman of the Committee on Research would present the annual report of that committee.

ANNUAL REPORT OF COMMITTEE ON RESEARCH—1956

[The following is abstracted from the full report of the committee which appears in the ASHAE JOURNAL SECTION, *Heating, Piping, & Air-Conditioning*, January 1957, pages 199 to 206.]

The Committee on Research which has overall responsibility for the Society's research activity is pleased to present here in brief the progress and plans made during the fiscal year, November 1, 1955 to October 31, 1956.

Ten projects at the Laboratory in Cleveland, Ohio and 10 projects at co-operating institutions were supported. Seven of the 10 projects at the Society's Laboratory were begun in 1956, replacing some completed during the previous year. Eleven technical papers were completed for publication.

The Technical Advisory Committees (TACs) and subcommittees held 28 meetings during the year, a new high in activity. The Committee on Research and Research Executive Committee met 8 times. The TACs are urged to suggest and evaluate opportunities for new research projects, and to review the progress of the laboratories. Their recommendations for new projects are carefully considered for activation by the Committee on Research. More than 275 engineers participate on the research committees.

EXPENDITURES AND FINANCES

In round figures, research expenditures totaled $214,000 for the year as compared with $205,000 in 1955. The receipts from contributions alone were $115,000 with the remainder of the support coming from income from dues and the research reserve allocated for laboratory use. The supporting companies, consultants and trade asso-

ciations who contributed funds numbered 201. Ten companies made gifts or loans of equipment. Acknowledgments of the vital participation of these groups is given with sincere thanks. The Committee on Research invites further participation in research from industry and from the profession.

The current staff at the laboratory consists of 6 engineers, 3 research assistants, 4 on the administrative staff, a librarian and 6 in the secretarial and maintenance personnel.

AIR CLEANING

University of Minnesota: In collaboration with the air-filter industry, the Society has continued its sponsorship of the study of air contaminants. A. B. Algren and K. T. Whitby have furthered the development of laboratory and field techniques for sampling and analyzing atmospheric dust, and for determining the dust-holding capabilities of air filters.

An improved photometer was produced and described for evaluating air cleaners by the dust-spot method. For accurate work glass fiber and millipore filters have been found to be superior to the chemical filter papers ordinarily used. Notes on the centrifuge size analysis of dust collected on millipore filters have also been exchanged with other investigators now using the centrifuge sedimentation method of dust analysis developed by the University.

A field survey of atmospheric dusts is in progress at 7 locations over the country, the samples from these locations will be analyzed for size distribution and other features.

A new dust feeder for testing air filters has been under development for better dispersion of test dusts than is possible with feeders now in use. Dispersion is to be obtained in an air nozzle where conditions of high turbulence and expansion exist. The dust survey and the feeder project are expected to lead to recommendations for test dusts that simulate the range of atmospheric particulate matter normally encountered in practice.

A study was also made of the basic physical phenomena by which small particles are trapped in fibrous filters.

AIR DISTRIBUTION

The prediction of flow patterns of air streams for grilles and nozzles is basic to satisfactory air conditioning design. Projects in this area were sponsored at 2 institutions.

Case Institute of Technology: Alfred Koestel and J. B. Austin, Jr., reported on an experimental study of 2 closely-spaced parallel air jets under isothermal conditions. The jets merged at now-predictable distances from the nozzles. Equations were presented to enable a designer to predetermine the velocities in the jet stream at various distances from the openings.

An investigation of air patterns from radial-flow outlets has been completed, as has a study of room heating by warm air at near 100 F. Reports are in preparation.

Kansas State College: Linn Helander and associates reported on the outlet characteristics that affect downthrow of heated air jets, such as from grilles and unit heaters. A simple and quick means was presented for measuring the thrust or projective force developed by outlets. Flow characteristics, velocity distribution downstream, and the influence of cores in the jet were given.

The project is being completed in 1957. A much better understanding of vertical air streams has resulted from the 5 papers issued to date.

The TAC on Air Distribution is considering the publication of a bulletin on air distribution in rooms, and new research on high-velocity air systems.

COMBUSTION

Combustion noise from residential boilers and warm-air furnaces heated by oil and gas burners is being investigated at Battelle Memorial Institute. Objectives are to

define the fundamental nature of the audible pulsations and to find methods for their suppression or complete elimination.

For the oil-fired unit, the important factors in oscillations appear to be (1) the relative flow patterns of combustion air, hot products of combustion, and fuel, and (2) the relative positions of the combustion zone and the pressure antinode. Motion pictures were taken to show the periodic, rapid, flame expansions.

For the gas-fired unit, the most important variable appears to be the rate of flow of primary air. The driving mechanism for the oscillations is related to vortex shedding of the flame.

Many variables in burner design and operation, such as spray angle of nozzles, burner tubes, fans and others have been investigated, as well as combustion space, length of gas travel, and location of barometric damper.

The project is continuing into 1957 with guidance from the Pulsations Steering Committee of the TAC on Combustion.

HEATING AND AIR-CONDITIONING LOADS

A major program is in progress by T. C. Min at the Research Laboratory to determine the infiltration through swinging-door entrances of commercial-type buildings during both winter-heating and summer-cooling regimes. Preliminary results confirm that the stack effect of tall buildings is governed almost directly by the building height and indoor-outdoor temperature difference. The time-motion cycle of door openings will be measured by motion pictures during normal traffic flow. Models of doors and vestibules have been employed in a wind tunnel to measure the air flow at various door angles. By combining the findings on draft, traffic, and door motions, it will be possible to formulate rules for estimating infiltration and their resultant effect on air-conditioning loads.

University of Illinois: Air infiltration of modern small residences, both single and two-story, has been investigated by the tracer-gas technique. The tests were run by injecting a quantity of helium into the house air and noting the decline in helium concentration as the air in the house was gradually diluted by outside air. The infiltration increased somewhat with wind velocity and with indoor-outdoor temperature difference, roughly doubling with a 50 F increase in temperature difference. The magnitude of infiltration ranged mainly between 0.2 and 0.6 air change per hour.

HEAT FLOW THROUGH FENESTRATION

The Society operates the only known solar calorimeter especially designed to measure the heat transfer through windows and skylights. The results are expressed directly in energy units of Btu per (square foot) (hour) or in percent of incident radiant energy which reaches the space beyond the inside surface. The effectiveness of awnings, Venetian blinds, shade screens, and other shading devices can also be determined with it.

After 10 years of use, the solar calorimeter was thoroughly overhauled and returned to working condition by September 1. Improvements were added to reduce future maintenance and to increase accuracy. Fortunately, favorable weather in abundance was available during September and October to conduct tests of certain shading devices.

Plans have been made to complete the study of awnings during the summer of 1957. Their effectiveness in reducing solar load through windows is known to be high but definite values are needed for analyses of the many variable factors entering into this problem.

Preparations are also under way to measure the solar heat gain next summer of double glazing with between-glass shading devices, such as Venetian blinds and shade screens. Consideration is also being given to tests of reflective surfaces behind glass, and to the new forms of clear and glass-fiber-reinforced plastic fenestrations.

The diffuse solar radiation striking inclined surfaces, such as roofs and skylights, will

be predictable for purposes of air-conditioning load calculation as the result of a study by A. N. Cerny nearing completion.

HOT WATER AND STEAM HEATING

The table on capacity of steam piping in THE GUIDE which has been widely used is based on both tests and field experience. However, it appears desirable occasionally to compare these values with present application practices to determine whether the factors of safety are still most representative. Such a study is now under way at the Research Laboratory with guidance from the TAC on Hot Water and Steam Heating.

University of Illinois: W. S. Harris is investigating the separation of dissolved gas from water in a one-pipe forced-circulation hot water heating system, and the entrainment of gas by water as it moves through the system. Both radiators and convectors have been used. More gas was formed at higher water temperatures, and more gas appears to separate from radiators than from convectors. As the return of gas to the compression tank is desirable, future tests propose locating the tank closer to the boiler and installing a dip tube in the boiler, with piping to the tank.

University of Florida: J. D. Simonds has been investigating air locks and air entrainment in hot water heating systems containing both radiators and convectors. Additional tests will be required before final conclusions can be reached.

Northwestern University: The test work on metastable state of water at boiling temperatures has been completed. The report will state the conditions under which the metastable state has been observed.

It is also planned to resume test work on noise in pipe systems during 1957.

INDUSTRIAL ENVIRONMENT

University of Pittsburgh: A basis for determining the requirements for exhaust ventilation of hot processes in industry has been sought for some time. The air streams flowing upward around a hot ingot or tank carry excess heat and often dust particles, fumes, and gases which contaminate the air in the plant.

The paper on *Air Flow in Free Convection Over Heated Bodies*, by T. F. Hatch and D. B. Barron-Oronzco, reports the results of a laboratory study. A general equation was produced to express the findings for practical design purposes. Exhaust-hood size and rate of exhaust ventilation can be calculated for a variety of conditions with allowance being made for cross drafts.

The University is currently engaged in a study of lateral exhaust ventilation with Society support.

INSULATION

THE GUIDE 1957 will contain the full fruits of the latest work of the TAC on Insulation. The newly-computed U values of overall-heat-transfer coefficients through building sections will be available in a more readily useable form for the benefit of architects and engineers, trade associations and manufacturers.

The TAC has recommended field and laboratory investigations of (1) perimeter insulation for slabs-on-grade, (2) loose fill insulation in walls, (3) crawl-space insulation, and (4) attic ventilation and insulation. The objective is to determine how conditions of use influence actual heat-flow magnitudes.

ODOR STUDIES

The paper on *Temperature and Humidity Effect on Odor Perception*, by Messrs. Kerka and Humphreys of the Society's Research Laboratory, reported revolutionary findings. Contrary to popular opinion, at a given concentration of odorant an increase in humidity has the definite effect of lowering the odor-perception level of cigarette smoke as well as of the pure odorant vapors tested.

Other factors remaining constant, a 10-fold increase in concentration of odorant in the air changes the perception from barely *noticeable* to *definite*, while a 100-fold increase

changes the perception from *noticeable* to *strong*. Exposure to an odor for as short as one minute caused a reduction in odor sensation equal to about 10-fold reduction in odor concentration.

Plans have been made with the TAC on Odors to continue the odor research with studies of odor adsorption and desorption by cotton, wool, and synthetic fabrics. Among the variables are odorant, temperature, humidity, time and ventilation. The Society's 2 odor test rooms have been equipped with automatic temperature and humidity control for that purpose. The adsorption and retention of odors by fabrics in room furnishings, namely rugs, upholstery and drapes, are important factors in odor control.

PANEL HEATING AND COOLING

The research in the completely paneled Environment Room resulted in 3 papers during the year.

A survey of plastic pipe usage, which was made, indicates that such pipe is under test by several organizations for use in heating concrete floor slabs. Satisfactory results are reported from installations up to 4 years old. Care must be taken to protect the pipe from rodents and to limit the upper water temperatures to about 160 F.

The overall program which began in 1949 will be completed with a paper on the heat transfer from fluorescent lighting fixtures to a cooled panel.

SENSATIONS OF COMFORT AND PHYSIOLOGICAL RESEARCH

The Comfort and Effective Temperature Charts of the Society have long served as an index to thermal environments. With the spread of air conditioning under a wide variety of conditions not fully explored at the time the charts were produced, additional information appears to be needed.

A major undertaking was launched during 1956 for a 10-year program of tests with human subjects on the effect of indoor thermal environment on comfort and human performance. The 12 x 24.5-ft Environment Room with temperature controlled walls, floor, and ceiling is being equipped with more flexible and extensive air-conditioning system to provide a wide range of temperatures, humidities, air motion and radiation.

Two air-conditioned pretest rooms are being constructed to provide a controlled environment for rest and to normalize body temperatures and clothing moisture before the tests. Adjacent to the test and pretest area an observation room in a central location is being provided for the research force. The control of the equipment to provide specified environment regime will be centralized in a control console.

Separate air-conditioning circuits for the Environment Room and pretest-observation areas have been designed for automatic and close control of air temperature, humidity, and motion, as well as of room surface temperatures.

The several projects to be undertaken over a period of years include:

1. Effect of humidity on comfort while subjects are (*a*) at rest, and (*b*) engaged in light activities.
2. Effect of ceiling and wall temperatures (radiation) on comfort while subjects are (*a*) at rest, and (*b*) engaged in light activities.
3. Optimum thermal environments for comfort while subjects are working at elevated rates of bodily heat output.
4. Effect of thermal environments on human performance and efficiency, such as alertness, learning, accuracy of work, fatigue and stress.

The detailed programs are being prepared by Nathaniel Glickman, consulting physiologist, with the cooperation of the TACs on Sensations of Comfort, Industrial Environment and Physiological Research.

SOLAR ENERGY COLLECTORS

University of Minnesota: Solar radiation is a promising source of energy for heating buildings and service water, as well as for absorption cooling systems. Of the many

problems involved in harnessing solar energy, two of the more important ones are (1) the extent and availability of the energy supply, and (2) the design and performance of flat-plate collectors.

The University has installed 2 such collectors, 7 ft-1 in. wide by 3 ft-3 in. high, with refrigeration systems and instrumentation for measuring collector performance under a variety of conditions. R. C. Jordan and J. L. Threlkeld have described the facilities in a recent paper to the Society. The experimental work is now under way and is expected to provide basic information over a period of years that will be applicable for a wide geographical area.

In the meantime the TAC on Solar Energy is preparing a report on *Solar Energy Utilization for Heating, Cooling, Distillation and Drying.*

SORPTION DEHUMIDIFICATION

A proposed test code on sorption dehumidifiers has been modified by the TAC on Sorption since receiving the test program report from The Pennsylvania State University.

The TAC has also recommended research to determine the effectiveness of the sorption dehumidification cycle on the destruction of micro-organisms in the air being dehumidified. Low humidities are known to destroy some bacteria in a rather short time. Certain temperature ranges during the regeneration of the sorbent are also lethal to bacteria. The overall sterilization under dynamic conditions needs to be determined. The project is scheduled for activation in 1957.

SOUND AND VIBRATION CONTROL

A practical, yet scientific, method for measuring the noise radiated by fans and allied air-conditioning equipment has been needed for a long time. To lay the foundation for a method that could be standardized, W. F. Kerka of the Research Laboratory has investigated the noise output of a 15-in. centrifugal fan by the four methods recommended by the TAC on Sound and Vibration Control:

1. Free-field test. The acoustic power radiated into a free space from the opening of the inlet and discharge ducts was determined by taking measurements around the end of the duct at 4 and 8 ft radii.
2. Reverberant-room test. The acoustic power radiated into a reverberant room from the duct openings was determined from measurements made in the space. The room was calibrated with a standard noise source of known acoustic output.
3. Duct-opening test. The acoustic power at the duct opening was found by taking measurements around the periphery of the duct opening, with and without air flow, and across the duct face at no flow. From these measurements, acoustic power at the duct opening with flow was found.
4. Duct test. The acoustic power transmitted down the duct was found by taking measurements in the airstream with a shielded microphone. A sound-absorbing termination was used at the end of the duct.

The decibel readings were taken for octave bands of sound frequencies over a total range of 20 to 9600 cps, and converted to watts of sound power for comparison. Correlated results were obtained by the reverberant-room and free-field methods, usable for unit heaters and other unducted equipment.

The technique of measuring of noise in a duct will be useful in the study of noise regeneration and attenuation in components of duct systems, to be undertaken in 1957.

THERMAL CIRCUITS

A systematic research procedure for dealing with problems in cyclic heat flow through building sections, as well as heat exchanges in rooms, has been amply demonstrated in several papers on thermal circuits. Many of the equations are exceedingly complex and

laborious. Messrs. Parmelee, Vance, and Cerny have demonstrated the use of the electronic differential analyzer for solving the problems in air conditioning with suitable accuracy. Other types of modern calculators and analogue computers are also useful.

The thermal-circuit technique has also been applied to heat-flow data for glass-block skylights. A forthcoming paper will analyze the heat flow and permit calculation of results for a variety of glass blocks.

The TAC on Thermal Circuits is considering a possible program to evaluate the thermal performance of building walls when the outside is subjected to solar radiation and the inside space is air conditioned.

University of California: H. A. Buchberg is expanding his thermal-circuit analysis of buildings and building sections to include control performance and wishes to extend the analyses to multi-room buildings.

WEATHER DATA

The TAC on Weather Data has adopted as its scope *the accumulation and editing of weather data for heating, ventilating, and air-conditioning design and operation.* In cooperation with the Weather Bureau and Air Force, new tabulations of winter and summer data are to be prepared for THE GUIDE. A report on winter design data is in preparation which will summarize the probable frequency of recurrence of given temperatures at selected locations.

Close collaboration is being maintained with committees of the *National Research Council of Canada* and the *Building Research Advisory Board.*

The next order of business was the receipt of the report of the Inspectors of Election, presented by Mr. Salinger as follows:

REPORT OF INSPECTORS OF ELECTION

Ballot for Officers:	Total
President, P. B. Gordon, New York, N. Y.	2379
First Vice President, E. R. Queer, University Park, Pa.	2379
Second Vice President, A. J. Hess, Los Angeles, Calif.	2379
Treasurer, C. H. Pesterfield, East Lansing, Mich.	2379
Members of Council (three-year term)	
W. G. Hole, Montreal, Que., Canada	2378
P. J. Marschall, Chicago, Ill.	2378
W. O. Stewart, Los Angeles, Calif.	2378
P. N. Vinther, Dallas, Tex.	2376
Committee on Research (three-year term)	
D. D'Eustachio, Port Allegany, Pa.	2379
E. S. Howarth, New Kensington, Pa.	2377
H. R. Limbacher, Kalamazoo, Mich.	2378
E. F. Snyder, Minneapolis, Minn.	2377
J. I. Trimble, Columbus, Ohio	2377
Committee on Research (one-year term)	
P. H. Yeomans, Philadelphia, Pa.	2378
TOTAL BALLOTS RECEIVED	2461
TOTAL LEGAL BALLOTS	2379
INVALID BALLOTS	82

Scattering votes: Members of Council 3 and Committee on Research 9.

Respectfully submitted,

R. J. SALINGER, *Chairman*
F. W. BRUNDAGE
L. C. PLAEHN

Chairman Gordon then turned the session over to Professor Jennings in order that the 4 technical papers on the program might be heard.

Following the close of the discussion period, Chairman Jennings turned the session back to 1st Vice President Gordon who declared the session adjourned at 12:10 p.m.

THIRD SESSION, TUESDAY, FEBRUARY 26, 9:30 A.M.

This session which took the form of a Symposium on Dehumidification was called to order by Treas. R. A. Sherman, Columbus, Ohio, in the Grand Ballroom, Conrad Hilton Hotel. He at once introduced John Everetts, Jr., Philadelphia, Pa., to act as Moderator.

Mr. Everetts outlined the nature of the program which consisted of 5 papers prepared by speakers selected to discuss major phases of the subject, as follows: I—*Dehumidification by Mechanical Refrigeration*—W. E. Ellis, Cincinnati, Ohio; II—*Dehumidification by Solid Sorbents*—W. L. Ross, Johnstown, Pa.; III—*Dehumidification by Liquid Sorbents*—E. W. Gifford, Milwaukee, Wis.; IV—*Economics of Dehumidified Storage*—G. W. Higgs, Jr., Washington, D. C.; and V—*Typical Dehumidification Problems and Their Solution*—E. R. Queer, University Park, Pa.

Successively as called on by Mr. Everetts, each author presented his prepared paper and the session was then thrown open for discussion. A written discussion had been prepared by G. W. Adolphson, Puget Sound Naval Shipyard, and the subject was orally discussed by H. E. Parker, Quincy, Mass., G. L. Simpson, Pittsburgh, Pa., G. C. F. Asker, Falls Church, Va., W. M. Viessman, Baltimore, Md., H. W. Lyons, Philadelphia, Pa., O. D. Colvin, New York, N. Y. and T. H. Urdahl, Washington, D. C. The questions raised and points discussed were referred by Moderator Everetts to the several speakers who further discussed and commented on them.

Since these symposium papers and the discussions are of an informal nature, it is not expected that they will be printed in full in TRANSACTIONS. However, the text of 2 of these papers, those by Dr. Ross and by Professor Queer, are printed in full in the ASHAE JOURNAL SECTION, *Heating, Piping & Air Conditioning*, March 1957, p. 161. Text of the remaining 3, those by Mr. Ellis, Mr. Gifford and Mr. Higgs, are published in the April 1957 issue of the JOURNAL SECTION, beginning on p. 152.

The text of all 5 of the papers presented at this Symposium is available from the Society in the form of a separately printed bulletin.

Following the close of the discussion period, Moderator Everetts turned the session over to Treasurer Sherman who adjourned it at 12:10 p.m.

FOURTH SESSION, WEDNESDAY, FEBRUARY 27, 9:30 A.M.

After opening the session in the Waldorf Room of the Conrad Hilton Hotel, First Vice President Gordon introduced successively the authors of 3 technical papers which were presented.

Following completion of the discussions, Chairman Gordon adjourned the session at 11:45 a.m.

FIFTH SESSION, WEDNESDAY, FEBRUARY 27, 9:30 A.M.

Second Vice Pres. E. R. Queer, University Park, Pa., opened the fifth session in the Grand Ballroom of the Conrad Hilton Hotel. This session took the form

of a Symposium on the general subject of Industrial Ventilation. Professor Queer at once introduced C. H. Pesterfield, East Lansing, Mich., who acted as moderator at the session.

Five papers had been arranged covering various topics as follows: I—*Control of the Industrial Environment*—P. J. Marschall, Chicago, Ill.; II—*Dust Particle Inertia and Various Consequences*—W. C. L. Hemeon, Pittsburgh, Pa.; III—*Contaminant Control*—K. E. Robinson, Detroit, Mich.; IV—*Summer-Winter Comfort Ventilation for Industry*—B. R. Small, Pittsburgh, Pa.; V—*Cleaning of Industrial Effluents*—J. M. Kane, Louisville, Ky.

Moderator Pesterfield called on these authors successively and they presented their prepared material after which the moderator indicated that questions from the floor could be directed to these various authors.

Because of the nature of these papers and of the discussions presented in connection with them, it is not expected that they will appear in the TRANSACTIONS. However, two of these papers, namely, paper II and paper IV are published in the February 1957 issue of the ASHAE JOURNAL SECTION, *Heating, Piping & Air Conditioning*. The paper II entitled Dust Particle Inertia and Various Consequences by Mr. Hemeon begins on p. 247 and the paper IV, Summer-Winter Comfort Ventilation for Industry, by Mr. Small, begins on p. 241. Paper V on the subject of Cleaning of Industrial Effluents, prepared by Mr. Kane, appears in the April 1957 JOURNAL SECTION, *Heating, Piping & Air Conditioning*, p. 166. Papers I and III are printed in the May 1957 issue of the ASHAE JOURNAL SECTION, beginning on page 155.

All 5 of the papers are available from the Society in a separately printed bulletin.

During the course of the discussion period, oral discussions were presented by V. J. Jamal, Niagara Falls, N. Y., J. J. Aeberly, Chicago, Ill., Warren Viessman, Baltimore, Md., J. H. Clarke, Chicago, Ill., I. S. Dane, Boston, Mass., H. W. Reinken, Detroit, Mich., W. A. Danielson, Memphis, Tenn., and K. T. Whitby, Minneapolis, Minn. Following the completion of the discussion period, Moderator Pesterfield turned the meeting back to Second Vice President Queer who adjourned the session at 12:00 p.m.

SIXTH SESSION, THURSDAY, FEBRUARY 28, 9:30 A.M.

Meeting in the Waldorf Room of the Conrad Hilton Hotel, Pres. John W. James, Chicago, Ill., called the sixth session to order and 3 technical papers were presented and discussed.

At this time, since the meeting was honored by the presence of the President of the *Institution of Heating and Ventilating Engineers* of England, J. Clifford Byles, President James invited him to address the session. In reply, Mr. Byles spoke briefly.

The next order of business was the Report of the Resolutions Committee and President James called on A. O. Roche, Jr., Indianapolis, Ind., who read the report of that committee which appears herewith.

RESOLUTIONS

WHEREAS, the 63rd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING EGINNEERS and the 13th International Heating and Air-Conditioning Exposition are now drawing to a close, and

Whereas, this meeting was held in Chicago where cooperation with the weatherman, after early signs of spring, finally made good its renowned claim to being the Windy City, and

WHEREAS, the Illinois Chapter and the Committee on Arrangements under the direction of Pete Marschall, general chairman, have ably provided for our comforts and entertainment and relaxation, and

WHEREAS, the arts and sciences of our profession have been advanced by the excellent choice of material for our Technical Sessions under the direction of the officers and staff of our Society, and

WHEREAS, the 13th International Heating and Air-Conditioning Exposition has broken all records for quality of exhibits, size and attendance; therefore

BE IT RESOLVED, that our appreciation and sincere thanks be extended:

TO the Illinois Chapter, its officers and committees who have made this meeting possible,

TO Pete Marschall, general chairman, Harry Gragg, vice-chairman and Illinois Chapter President, and to Honorary Chairmen, H. M. Hart, S. R. Lewis and Dr. A. C. Willard,

TO Mrs Harry Gragg and the ladies of her committee who so outstandingly provided for the welfare and entertainment of our visiting Ladies,

TO Dr. Richard Heflebower for an interesting preview of things to come,

TO the authors and discussers of the technical papers and symposiums for their educational contributions,

TO the Chicago Hotels for providing for our needs and comforts,

TO Bill Kuechenberg who so ably presided at the Annual Banquet,

TO Dr. Preston A. Bradley, who in 4 short years has been promoted from our luncheon speaker to our banquet speaker, for his inspirational address on *Facing the Future.*

TO Maurice K. Fahnestock for his many years of service and creative contributions to our Society and the profession, who has been presented the F. Paul Anderson Award in recognition of these achievements.

TO E. K. Stevens and the International Exposition Co. for the excellent presentation of the products of our industry.

TO Paul M. Chung for his contributions to the arts for which he was chosen the first recipient of the ASHAE-Homer Addams Award.

TO the members of our Society who have been honored with the new membership grade of Fellow. These distinguished gentlemen are: Walter L. Fleisher, L. N. Hunter, J. Donald Kroeker, C. S. Leopold, Samuel R. Lewis.

TO President John W. James who this Resolutions Committee feels has been outstanding in adding dignity and stature to this office throughout the world, and

TO the other Officers, Committees, Staff and all others who have so generously given of their time and efforts which has resulted in this successful meeting.

Respectfully submitted

The Resolutions Committee

A. O. ROCHE, JR., *Chairman*, Indianapolis, Ind.
A. W. EDWARDS, Cincinnati, Ohio
J. W. MAY, Louisville, Ky.

Following adoption of the resolutions and since there was no unfinished business or new business to come before the session, President James adjourned the Meeting at 12:15 p.m.

PROGRAM—63rd ANNUAL MEETING

Conrad Hilton Hotel, Chicago, Ill.
February 25-28, 1957

Saturday—February 23

9:00 a.m. Membership Committee, B. W. Farnes, *Chairman* (*Room 9*)
10:00 a.m. Executive Committee, P. B. Gordon, *Chairman* (*Room 5*)
2:00 p.m. Finance Committee, A. J. Hess, *Chairman* (*Room 5*)
2:00 p.m. Regions Central Committee, E. R. Queer, *Chairman* (*Room 9*)
2:00 p.m. Research Executive Committee, B. H. Jennings, *Chairman* (*Room 19*)

Sunday—February 24

8:00 a.m. Charter and By-Laws Committee, A. E. Stacey, Jr., *Chairman*
10:00 a.m. REGISTRATION (*Beverly Room*)
10:00 a.m. Council Meeting (*Lipton Room—Stock Yards Inn*)
10:00 a.m. Committee on Research, B. H. Jennings, *Chairman* (*Room 19*)
3:00 p.m. Reception and Refreshments with a Hawaiian accent (*Bel Air Room*)

Monday—February 25

8:00 a.m. Authors' Breakfast (*Park Row*)
9:00 a.m. REGISTRATION (*Normandie Lounge*)
HOSPITALITY HEADQUARTERS (*Normandie Lounge*)
9:30 a.m. FIRST SESSION (*Waldorf Room*)

Call to Order—Pres. John W. James

Greetings by P. J. Marschall, *Chairman* Committee on Arrangements

Report of President—John W. James

Report of Council—A. V. Hutchinson

Report of Treasurer—R. A. Sherman

Study of Three Redwood Cooling Tower Packings, by W. W. Smith, College Station, Texas, presented by Mr. Smith

Earth as Heat Source and Sink for Heat Pumps, by D. M. Vestal, Jr. and B. J. Fluker, College Station, Texas, presented by Mr. Fluker

From '36 to '56: Air Conditioning Comes of Age, by Walter A. Grant, Syracuse, N. Y., presented by Mr. Grant

12:00 noon WELCOME LUNCHEON (*Grand Ballroom*)

Toastmaster: C. M. Burnam, Jr.

Speaker: Dr. Richard Heflebower, *Chairman*, Economics Dept., Northwestern University

Subject: Revolutions In Our Times—What Our Lives Will Be Like in the Next 10 Years.

Presentation of Certificates to Fellows by Pres. John W. James: W. L. Fleisher, J. Donald Kroeker, L. N. Hunter, C. S. Leopold, S. R. Lewis

Presentation of ASHAE-Homer Addams Award by Paul K. Addams to Paul M. Chung, Graduate Student, University of Minnesota

2:00 p.m. EXPOSITION—Opening of 13th International Heating and Air Conditioning Exposition (*International Amphitheatre*). ASHAE Booth 701

2:00 p.m. Guide Advisory Committee, C. E. Price, *Chairman* (*Room 12*)
2:15 p.m. TAC on Air Cleaning, E. F. Snyder, Jr., *Chairman* (*Room 3*)
2:15 p.m. TAC on Air Distribution, W. O. Huebner, *Chairman* (*Room 4*)
2:15 p.m. TAC on Hot Water and Steam Heating, H. A. Lockhart, *Chairman* (*Room 2*)
2:15 p.m. TAC on Solar Energy Utilization, R. C. Jordan, *Chairman* (*Room 1*)
2:15 p.m. TAC on Weather Data, John Everetts, Jr., *Chairman* (*Room 9*)
2:30 p.m. Ladies' Talent and Tea (*Crystal Ballroom*, Sheraton-Blackstone Hotel)
9:00 p.m. Get-Together Party (*Grand Ballroom*)

Tuesday—February 26

8:00 a.m. Nuclear Energy Engineering Committee, R. A. Sherman, *Chairman* (*Park Row*)
8:00 a.m. Authors' Breakfast (*Park Row*)
9:00 a.m. REGISTRATION (*Normandie Lounge*)
HOSPITALITY HEADQUARTERS (*Normandie Lounge*)
9:30 a.m. SECOND SESSION (*Waldorf Room*)
Call to Order—1st Vice Pres. P. B. Gordon
Report of Committee on Research—B. H. Jennings, *Chairman*
Revised Winter Outside Design Temperatures, by H. C. S. Thom, Washington, D. C., presented by Mr. Thom
Analysis of an Air-Conditioning Thermal Circuit by an Electronic Differential Analyzer, by G. V. Parmelee, Dhahran, Saudi Arabia, P. Vance, Cambridge, Mass., and A. N. Cerny, Cleveland, Ohio, (by title only)
Pulsations in Residential Heating Equipment—Preliminary Results, by A. A. Putnam and W. R. Dennis, Columbus, Ohio, presented by Mr. Putnam
The Dust Spot Method for Evaluating Air Cleaners, by K. T. Whitby, A. B. Algren and R. C. Jordan, Minneapolis, Minn., presented by Mr. Whitby
Report of Inspectors of Election
9:30 a.m. THIRD SESSION (*Grand Ballroom*)
Call to Order—Treas. R. A. Sherman
SYMPOSIUM ON DEHUMIDIFICATION
John Everetts, Jr., *Moderator*
Dehumidification by Mechanical Refrigeration—W. E. Ellis
Dehumidification by Solid Adsorbents—W. L. Ross
Dehumidification by Liquid Sorbents—E. W. Gifford
Economics of Dehumidified Storage—G. W. Higgs, Jr.
Typical Dehumidification Problems and Their Solution—E. R. Queer
12:00 noon EXPOSITION (*International Amphitheatre*)
12:15 p.m. Life Members' Dutch Treat Luncheon (*Parlor A*)
1:30 p.m. Nominating Committee Meeting (*Room 2*)
2:00 p.m. Guide Committee, M. W. Keyes, *Chairman* (*Room 5*)
2:15 p.m. TAC on the Heat Pump, F. R. Ellenberger, *Chairman* (*Room 13*)
2:15 p.m. Joint meeting of TAC on Industrial Environment, P. J. Marschall, *Chairman* and TAC on Physiological Research, M. K. Fahnestock, *Chairman* (*Room 4*)

2:15 p.m. TAC on Sorption, G. L. Simpson, *Chairman* (*Room 14*)
2:15 p.m. TAC on Sound and Vibration Control, H. A. Lockhart, *Chairman* (*Room 18*)
3:30 p.m. ASHAE-*ASRE* Committee on Cooperation (*Room 9*)
7:00 p.m. Past Presidents' Dinner (*Room 3*)
7:30 p.m. *ASA* Sub-Committee on Sound (*Room 19*)
7:30 p.m. TAC on Heat Transfer Through Fenestration, R. W. McKinley, *Chairman* (*Room 13*)
7:30 p.m. TAC on Heating and Air Conditioning Loads, H. T. Gilkey, *Chairman* (*Room 14*)
7:30 p.m. TAC on Insulation, M. W. Keyes, *Chairman* (*Room 4*)
7:30 p.m. TAC on Odors, A. B. Hubbard, *Chairman* (*Room 18*)
7:30 p.m. TAC on Plant and Animal Husbandry, A. J. Hess, *Chairman* (*Room 5*)
7:30 p.m. TAC on Thermal Circuits, S. F. Gilman, *Chairman* (*Room 12*)

Wednesday—February 27

8:00 a.m. Authors' Breakfast (*Park Row*)
9:00 a.m. REGISTRATION (*Normandie Lounge*)
HOSPITALITY HEADQUARTERS (*Normandie Lounge*)
9:30 a.m. FOURTH SESSION (*Waldorf Room*)
Call to Order—1st Vice Pres. P. B. Gordon
Cold Wall Effects in a Ceiling-Panel-Heated Room, by L. F. Schutrum and T. C. Min, Cleveland, Ohio, presented by Mr. Schutrum
Performance of Covered Hot Water Floor Panels, Part II—Room Conditions, by E. L. Sartain, Racine, Wis., and W. S. Harris, Urbana, Ill., presented by Mr. Sartain
Thermal Design of Warm Water Concrete Floor Panels—A Design Manual prepared by a subcommittee of TAC on Panel Heating and Cooling and members of the staff of the ASHAE Research Laboratory, presented by L. C. Schutrum, Cleveland, Ohio
9:30 a.m. FIFTH SESSION (*Grand Ballroom*)
Call to Order—2nd Vice Pres. E. R. Queer
SYMPOSIUM ON INDUSTRIAL VENTILATION
C. H. Pesterfield, *Moderator*
Control of the Industrial Environment—P. J. Marschall
Contaminant Control—K. E. Robinson
Cleaning of Industrial Effluents—J. M. Kane
Summer-Winter Comfort Ventilation for Industry—B. R. Small
Dust Particle Inertia and Various Consequences—W. C. L. Hemeon
12:00 noon EXPOSITION (*International Amphitheatre*)
12:30 p.m. Ladies' Luncheon and Entertainment (South Shore Country Club)
1:30 p.m. Chapter Editors' Conference, J. H. Fox, *Chairman* (*Room 14*)
1:30 p.m. Publication Committee, G. B. Priester, *Chairman* (*Room 19*)
2:00 p.m. Committee on Ventilation Standards, Albert Giannini, *Chairman* (*Room 5*)
2:00 p.m. Organization Meeting of 1957 Committee on Research (*Room 1*)
2:30 p.m. Inspection Tour of Prudential Plaza
3:30 p.m. Program and Papers Committee, W. A. Grant, *Chairman* (*Room 18*)
6:30 p.m. President's Reception (*Beverly Room*)

6:30 p.m. Social Hour (*Normandie Lounge*)

7:00 p.m. ANNUAL BANQUET (*Grand Ballroom*)

Toastmaster: W. A. Kuechenberg

Speaker: Dr. Preston A. Bradley, Pastor of Peoples Church, Chicago

Subject: Facing the Future

Introduction and Installation of 1957 Officers

Presentation of Past President's Award to John W. James

Presentation of F. Paul Anderson Medal to M. K. Fahnestock

10:00 p.m. Dancing

Thursday—February 28

8:00 a.m. Authors' Breakfast (*Park Row*)

9:00 a.m. REGISTRATION (*Normandie Lounge*)

9:30 a.m. SIXTH SESSION (*Waldorf Room*)

Call to Order—Pres. John W. James

Outlet Characteristics That Affect the Downthrow of Heated Air Jets, by Linn Helander, Shee-Mang Yen and Wilson Tripp, Manhattan, Kansas, presented by Mr. Helander

Air Flow in Free Convection over Heated Bodies, by T. F. Hatch and D. Barron-Oronzco, Pittsburgh, Penna., presented by Mr. Hatch

Radioactive Process Ventilation, by S. H. Glassmire and J. P. Wahlen, Los Alamos, N. M., presented by Mr. Glassmire

Report of Resolutions Committee

Unfinished Business

New Business

Adjournment

12:00 noon EXPOSITION (*International Amphitheatre*)

1:00 p.m. Organization Meeting of 1957 Council (*Room 3*)

COMMITTEE ON ARRANGEMENTS

P. J. Marschall, *General Chairman*

H. G. Gragg, *Vice Chairman*

Honorary Chairmen

H. M. Hart S. R. Lewis Dr. A. C. Willard

RESERVATIONS: G. G. Freyder, *Chairman*

Banquet—G. V. Zintel, *Co-Chairman*; E. B. Boston, W. H. Brackett, Joseph David, N. A. Gregornik, W. C. Kadow, R. W. Reed, I. R. Saunders, M. S. Simon, D. H. Snyder, J. A. Solon, A. G. Sutcliffe, J. F. Tobin.

Get-Together Party—L. H. Streb, *Co-Chairman*; C. O. Anderson, Thomas Cranage, D. W. Davis, Jr., C. L. Meyer.

Welcome Luncheon—C. M. Burnam, Jr., *Co-Chairman*; A. H. Blaker, H. E. Clo, S. F. Rost, Harvey Schoeman.

Ladies' Activities—M. J. Bamond, *Co-Chairman*; R. J. Abramson, L. A. Barron, J. A. Holbrook, W. H. Kramer, A. W. Lippitt, R. W. Van Valzah.

Exposition—W. G. Pennington, *Co-Chairman*; H. F. Abendroth, R. T. Flickinger, J. J. Friedler, Jr., John Kastholm, Jr., L. P. Lang, J. W. Muessig, L. E. Schulein, J. T. Wells.

ENTERTAINMENT: J. S. Kearney, *Chairman*

Orchestra, Acts, etc.—H. E. Anderson, *Co-Chairman*; W. J. Abraham, J. J. Floreth, C. O. Pratt, W. A. Stahl.

Ladies Activities—Mrs. H. G. Gragg, *Co-Chairman*; Mmes. H. E. Anderson, M. J. Bamond, M. W. Bishop, C. M. Burnam, Jr., H. J. Couch, G. G. Freyder, E. P. Heckel, Jr., J. S. Kearney, Herbert Kreisman, W. A. Kuechenberg, P. J. Marschall, L. O. Paul, W. G. Pennington, R. W. Roose, R. J. Salinger, J. C. Scott, H. C. Stevens, L. H. Streb, E. R. Teske, G. V. Zintel.

HOSPITALITY: Herbert Kreisman, *Chairman*

Reception—M. W. Bishop, *Co-Chairman*; R. D. Anderson, A. M. Bladen, G. W. Bornquist, B. L. Casey, H. G. Chapin, Melvin Fakter, J. M. Frank, G. D. Haberer, Louis Hatz, B. H. Jennings, S. R. Lewis, F. W. Luft, Jr., E. N. Patek, S. A. Pope, William Shelley, R. W. Stewart, W. T. Walters.

Registration—R. J. Salinger, *Co-Chairman*; A. B. Allen, M. B. Chalfen, M. E. Coobs, T. H. Cloke, Jr., O. P. Crose, R. N. Demir, G. R. Healy, G. A. Mansinger, F. W. Riederer.

Inspection Trips—J. C. Scott, *Co-Chairman*; N. E. Bueter, S. H. Rich.

Information—E. R. Teske, *Co-Chairman*; J. B. Hill, S. M. Miner, J. J. Phillipi, Henry Seetch, Jr.

Transportation—H. C. Stevens, *Co-Chairman*; S. O. Johnson, J. H. Lazar.

Finance—E. P. Heckel, Jr., *Chairman*; C. W. Blodgett, K. G. Carlson, E. I. Kadlec, Jr., S. C. Tarnoff, C. M. Vreuls.

SESSIONS: L. O. Paul, *Chairman*

Arthur Abrahamson, M. K. Arenberg, M. M. Hattis, C. P. Iseminger, B. A. Krulewitch, H. T. Kucera, W. S. Maxim, J. G. Pecis, D. K. Ridley, J. E. Sorensen, C. E. Tomlin, W. W. Treichler, Jr., A. R. Wallace.

PUBLICITY: R. W. Roose, *Chairman*

L. A. Erickson, E. H. Gage, J. H. Gazlay, T. V. Johnson, W. L. Lowry.

FACILITIES: H. J. Couch, *Chairman*

C. B. Bendix, A. C. De Smet, Edmund Gelfand, E. H. Gustafson, E. M. Kralovec, Jr., L. P. Lebow, Eugene Rovin, A. R. Thompson, Jr., E. J. Williams.

No. 1591

STUDY OF THREE REDWOOD COOLING TOWER PACKINGS

By WESLEY W. SMITH*, COLLEGE STATION, TEX.

THREE types of redwood cooling tower packings were subjected to experimentation in efforts toward 3 objectives: (1) to obtain information of a fundamental nature on mass transfer and on resistance to air flow in various cooling tower packings; (2) to determine why individual cooling tower manufacturers have individual preferences as to packing types; and (3) to point the way toward improvement in packings.

The experimentation was accomplished in a 2- by 2-ft counterflow section which can handle packing arrangements up to 9 ft in height. It is an indoor installation specifically designed for evaluating[1] packing and other cooling tower components (see Fig. 1).

Several hundred tests performed with this equipment over a period of more than a year are the foundation for the following presentation.

DESCRIPTION OF PACKING AND DETAILS OF PROBLEM

In an effort to obtain as much information as possible on many of the variables known to affect cooling tower performance without creating an impasse of complexity, the testing was planned for 3 readily prepared and related packing styles each to be used with 3 vertical spacings and 3 thicknesses. Water flow rates ranging from approximately 1½ to 4½ gpm per sq ft and air velocities of from 275 to 550 fpm were considered as reasonable and easily obtained. Manual control of the water-on temperature at approximately 105 F was considered feasible but control of the wet-bulb and water-off temperatures too difficult and time consuming with the present equipment.

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* Associate Research Engineer, Texas Engineering Experiment Station, Texas A. & M. College System.

[1] A Test Section for Evaluating Cooling Tower Components, by A. B. Alter (M.S. Thesis, A. & M. College of Texas, College Station, Tex., May 1953).

Presented at the 63rd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Chicago, February 1957.

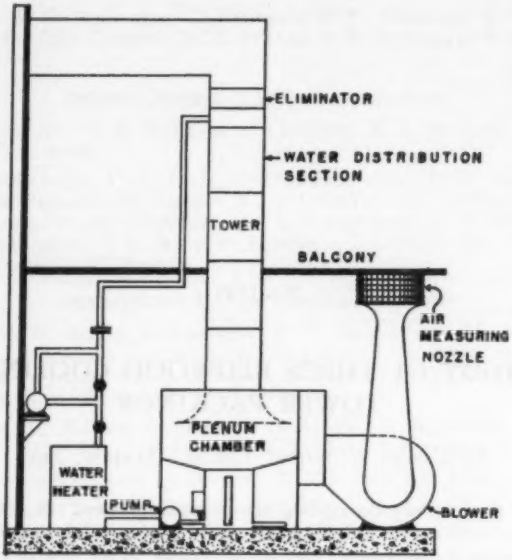


FIG. 1—FACILITY FOR 2- BY 2-FT COUNTERFLOW TESTS

TABLE 1—DETAILS OF PACKINGS AND PERFORMANCE DATA

ITEM NO.		TOWER PACKINGS TESTED		
		FIRST TYPE	SECOND TYPE	THIRD TYPE
1	Ratio of packing slat cross-section—width/height, (M)	0.36	1.00	2.78
2	Slat width, inches	0.36	0.59	1.00
3	Slat height, inches	1.00	0.59	0.36
4	Slat cross-sectional area, sq in.	0.36	0.35	0.36
5	Number of slats per deck	21.	13.	8.
6	Projected area, (sq ft), per deck per sq ft of tower ground area	0.298	0.306	0.316
7	Board ft of material per deck per sq ft of tower ground area	0.298	0.181	0.113
8	Surface area, (sq ft), per deck per sq ft of tower ground area	2.26	1.21	0.86
9	Volume coefficient of mass transfer, U^a, (for $N = 1$; $L = 1250$; and $G = 1250$)	130.4	106.2	86.6
10	U per board ft (Item 9 ÷ Item 7)	438.	587.	765.
11	U per sq ft proj. area (Item 9 ÷ Item 6)	438.	347.	274.
12	U per sq ft surf. area (Item 9 ÷ Item 8)	57.7	87.8	100.6
13	P, (pressure drop, inches water, per ft of height (for $N = 1$; $L = 1250$; $G = 1250$)	0.00681	0.00690	0.00954
14	P, per board foot (Item 13 ÷ Item 7)	0.0228	0.0381	0.0843
15	P/U (Item 13 ÷ Item 9)	0.522×10^{-4}	0.650×10^{-4}	1.102×10^{-4}

[a] The value of the coefficient U was obtained from Equation 3.

Fig. 2 and Table 1 show the details of the first packing tested. This 2-ft square packing unit made of $\frac{3}{8} \times 1$ in. nominal dimension redwood slats placed on edge is a standard packing used by 2 large manufacturers of cooling towers.

The second type (Table 1) was made from 13 square slats each having a cross-sectional area of 0.35 sq in. spaced to maintain essential constancy of projected area.

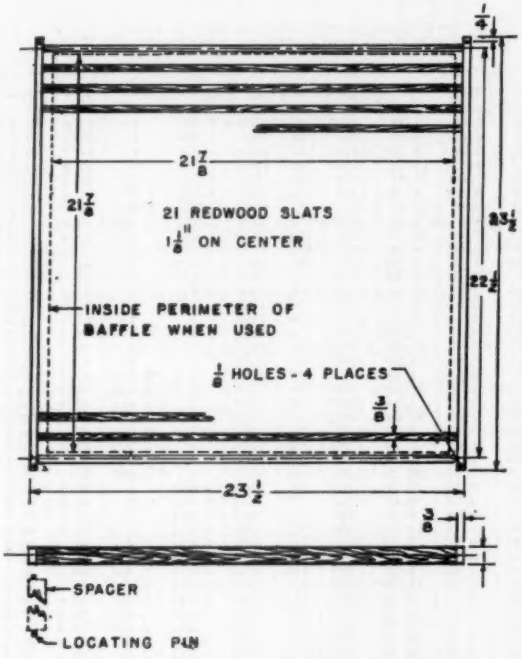


FIG. 2—DETAILS OF FIRST PACKING TESTED

The third type of packing tested (Table 1) was constructed of 8 of the same slats but arranged flatwise and the smaller number spaced to give essentially the same projected area as in the first type.

The 2 side pieces were $\frac{3}{8} \times 1$ in. for all the packing units.

It should be noted that the slats used in all the packings have essentially the same cross-sectional area and that all packing units have essentially the same projected area. The number of slats and the amount of total surface area vary with the ratio, M, of slat width to height.

The mass transfer per cubic foot of tower volume per unit vapor content potential was taken as the criterion of performance and was obtained from the correlation of the results of numerical integration of the commonly used cooling tower equation:[2]

$$UV/L = \int_{T_2}^{T_1} \frac{dT}{h'' - h_a} \qquad (1)$$

TABLE 2—TYPICAL DATA OBTAINED IN TESTS

TEST NO.	L/G	UV/L	V	L	U^a	G	$M^{-0.2b}$	$N^{0.8}$	$L^{0.8}$	$G^{0.8}$	U'^c	N	Z
1026	1.50	0.48	5.11	2288	215	1526	1.227	1	47.8	39.1	195	1	3
1028	0.51	0.63	5.11	814	100	1596	1.227	1	28.5	40.9	119	1	3
1031	0.86	0.88	8.11	1388	151	1606	1.227	1	37.2	40.1	155	1	6
1034	0.52	1.52	11.11	839	115	1603	1.227	1	29.0	40.0	121	1	9
1036	1.43	1.00	11.11	2278	205	1597	1.227	1	47.7	40.0	199	1	9
078	0.87	0.69	4.49	1395	214	1607	1.227	1.414	37.4	40.1	221	2	3
980	1.41	0.92	7.49	2291	281	1625	1.227	1.414	47.9	40.3	284	2	6
982	0.54	1.51	7.49	878	177	1637	1.227	1.414	29.6	40.5	177	2	6
984	0.85	1.69	10.49	1391	224	1638	1.227	1.414	37.3	40.5	223	2	9
1019	0.88	0.89	4.06	1394	306	1591	1.227	2	37.4	39.9	311	4	3
1014	1.41	1.26	7.06	2273	405	1611	1.227	2	47.7	40.1	398	4	6
1016	0.55	2.10	7.06	874	260	1590	1.277	2	29.6	39.9	246	4	6
1002	0.86	2.17	10.06	1389	300	1606	1.227	2	37.3	40.1	312	4	9
1073	0.54	0.62	5.59	861	96	1587	1	1	29.3	39.8	99	1	3
1075	1.43	0.44	5.59	2278	179	1596	1	1	47.7	40.0	162	1	3
1077	0.87	0.78	8.59	1390	126	1601	1	1	37.3	40.0	127	1	6
1080	0.55	1.27	11.59	883	97	1604	1	1	29.7	40.0	101	1	9
1082	1.41	0.90	11.59	2260	175	1599	1	1	47.5	40.0	162	1	9
1085	0.87	0.63	4.83	1394	182	1606	1	1.414	37.3	40.1	180	2	3
1087	0.52	1.20	7.83	839	129	1605	1	1.414	29.0	40.1	140	2	6
1089	1.44	0.82	7.83	2281	239	1589	1	1.414	47.8	39.9	229	2	6
1092	0.85	1.38	10.83	1394	178	1633	1	1.414	37.3	40.4	181	2	9
1096	0.86	0.82	4.29	1389	265	1608	1	2.000	37.3	40.1	254	4	3
1098	1.42	1.01	7.29	2280	316	1609	1	2.000	47.8	40.1	326	4	6
1100	0.54	1.74	7.29	873	208	1606	1	2.000	29.5	40.1	203	4	6
1103	0.87	1.81	10.29	1386	244	1595	1	2.000	37.2	39.9	252	4	9
1047	0.84	0.50	6.18	1350	109	1606	0.814	1	36.7	40.1	101	1	3
1042	1.43	0.64	9.18	2283	159	1594	0.814	1	47.8	39.9	132	1	6
1044	0.55	0.95	9.18	881	91	1592	0.814	1	29.7	39.9	82	1	6
1039	0.88	1.11	12.18	1400	127	1597	0.814	1	37.4	40.0	103	1	9
1051	0.87	0.57	5.25	1385	150	1596	0.814	1.414	37.2	40.0	146	2	3
1054	1.40	0.76	8.25	2281	210	1627	0.814	1.414	47.8	40.3	188	2	6
1056	0.55	1.06	8.25	883	114	1610	0.814	1.414	29.7	40.1	116	2	6
1059	0.87	1.23	11.25	1388	152	1592	0.814	1.414	37.2	39.9	145	2	9
1062	1.44	0.55	4.59	2283	273	1582	0.814	2.00	47.8	39.8	263	4	3
1064	0.53	0.87	4.59	839	159	1588	0.814	2.00	29.0	39.8	160	4	3
1066	0.88	1.17	7.59	1389	214	1582	0.814	2.00	37.3	39.8	205	4	6
1069	1.41	1.15	10.59	2288	249	1619	0.814	2.00	47.8	40.2	266	4	9
1071	0.53	1.68	10.59	839	133	1586	0.814	2.00	29.0	39.8	160	4	9

[a] Obtained from test and integral.

[b] When $M^{-0.2}$ is 1, $M = 1$; when $M^{-0.2}$ is 0.814, $M = 2.78$; when $M^{-0.2}$ is 1.227, $M = 0.36$.

[c] Obtained from $U' = 0.085\, M^{-0.2}\, N^{0.8}\, L^{0.8}\, G^{0.8}$ (Equation 3).

Measuring of the makeup water was eliminated after the initial tests as values required for the evaluation of Equation 1 could be obtained without it.

The initial pressure drop measurements were obtained by the subtraction of the positive pressure in the plenum chamber with an unpacked tower from that with

[2] HEATING, VENTILATING, AIR-CONDITIONING GUIDE, 1952, p. 760.

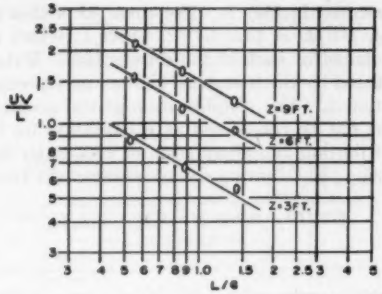


FIG. 3—TOWER TRANSFER CHARACTERISTICS

the subject packing. As the resistance to air flow through the eliminator and other parts of the tower was large compared to the resistance in the packing, the initial measurements were not entirely satisfactory. Therefore a special total pressure pickup device was installed resulting in more consistent results.

MASS TRANSFER COEFFICIENT

To illustrate the general method used in the following development and to give some idea of the degree of approximation involved, typical test values of the dimensionless group, UV/L, with the corresponding values of the dimensionless

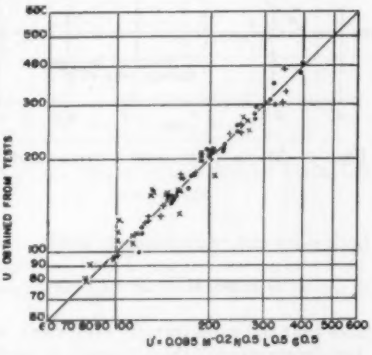


FIG. 4—DEVIATION OF MASS TRANSFER TEST DATA FROM VALUES GIVEN BY THE DERIVED EQUATION

group, L/G, are presented in Fig. 3. The three lines shown are represented by the general equation, $UV/L = C(L/G)^{-0.5}$, where C, which is also dimensionless, has a definite fixed value for each of the drawn lines. Values for C are entirely empirical, being dictated by the test data, and are an intermediate step in the development of Equation 2. The simplification of the problem resulting from the approximation might not be considered as justification for its use if it were not that even water distribution and elimination of wall water were not attained. It was possible to develop the following equations from the test data.

$$UV/L = 0.085\ M^{-0.2}\ N^{0.5}\ V(L/G)^{-0.5} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

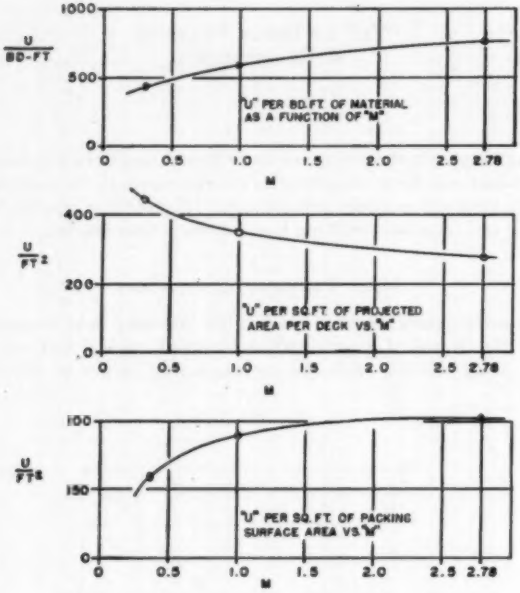


FIG. 5—MASS TRANSFER COMPARISONS

Eliminating V from both sides and transposing L yields

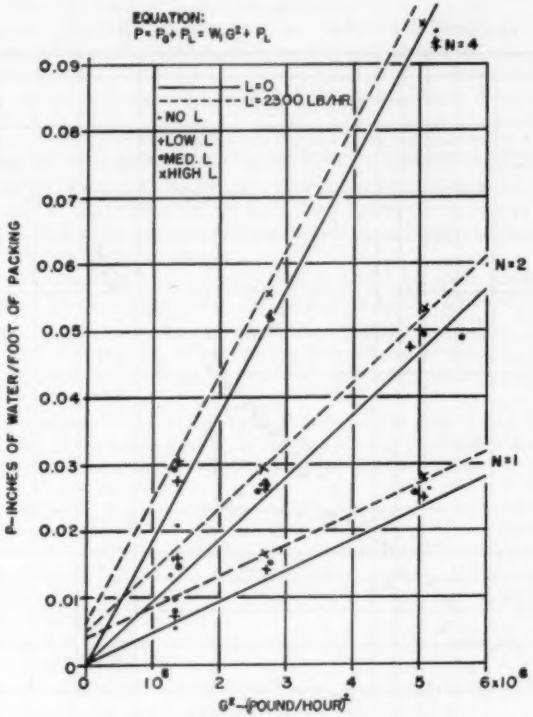
$$U = 0.085\ M^{-0.2}\ N^{0.5}\ L^{0.5}\ G^{0.5} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

This Equation 3 states that the mass transfer coefficient, U, increases as the 0.5 power of L, G and N, but decreases with the 0.2 power of the ratio of width to depth of slat cross-section.

All values of the coefficient U obtained from test and Equation 1 are shown in Fig. 4 with the corresponding values obtained from Equation 3. The agreement

is considered good in view of the complexity of the problem and the known deficiencies of control.

For the conditions $L = G = 1250$ lb per hr and $N = 1$, the value for U per board foot of packing material is presented in Fig. 5 as a function of M as is also U per



(Figure shows pressure drop per foot of packing for the tests when $M = 2.78$)

FIG. 6—TYPICAL VALUES OF PRESSURE DROP OBTAINED BY TESTS

sq ft of projected area per deck and U per sq ft of surface area. Table 1 contains the values just mentioned and also the important physical dimensions on the 3 packing styles.

PRESSURE DROP

Resistance proved to be essentially linear with packing height, and therefore few measurements were taken except for the maximum number of sections. Calibra-

tion pressure drop values obtained for the facility with no sections were subtracted from the 3 section manometer readings before calculating P. Since the resistance with no water flow appeared to be very nearly proportional to the flow indicating manometer reading, it was plotted as a function of G^2. Fig. 6 shows typical values of pressure drop obtained by test. The additional resistance caused by the water

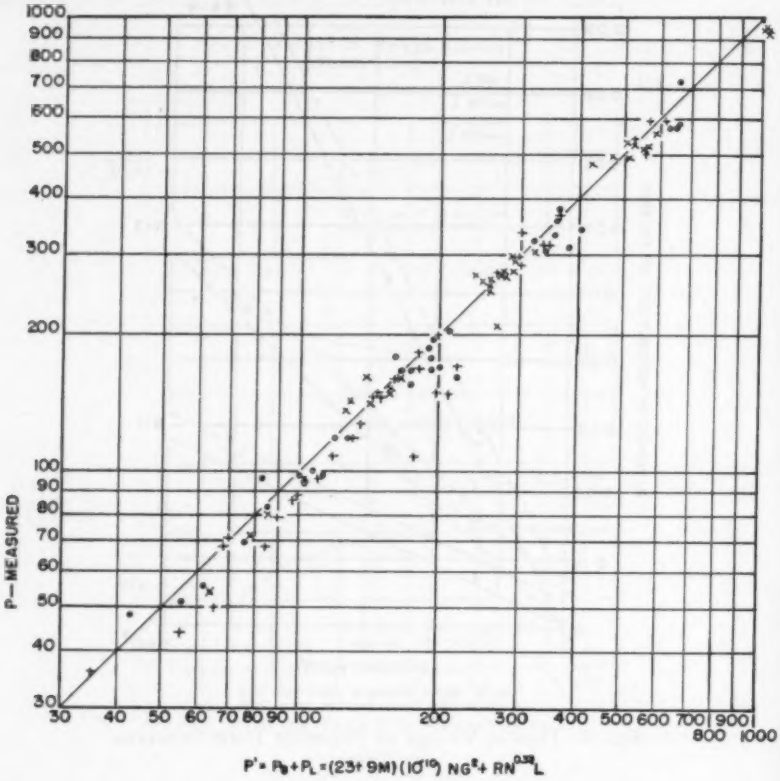


FIG. 7—DEVIATION OF PRESSURE LOSS TEST DATA FROM VALUES GIVEN BY THE DERIVED EQUATION

is represented as a constant value for any packing configuration and air flow and for simplicity is considered linear with water flow. The solid lines represent the no-water resistance, the dotted lines the resistance with 2300 lb of water per hr per sq ft of tower cross section.

An analysis of these data led to the development of an equation for the resistance to air flow as a function of the packing style, spacing amount, and rates:

$$P = P_G + P_L = (23 + 9M)\ (10^{-10})\ NG^2 + RN^{0.33}L \quad . \quad . \quad . \quad . \quad (4)$$

Note that R takes on the value of 2.17×10^{-6}, 1.52×10^{-6}, and 1.65×10^{-6} respectively for M values of 0.36, 1.00, and 2.78.

All measured values for pressure drop are shown as ordinates in Fig. 7 with the corresponding values calculated from Equation 4 as abscissas.

COMPARATIVE COSTS

As in so many engineering problems, the final comparison must be made on an economic basis in order to evaluate properly both the investment costs and operating costs. However, since each cooling tower installation has its own set of economic variables—cost of materials, cost of land, local restrictions, cost of labor, cost of power, water treatment and availability, conditions of operation, and others, no formal treatment will be attempted.

Nevertheless, comparative cost calculations based on what are believed to be representative values for the many variables lead to the following statements:

1. The cost of the tower structure itself, the accessory equipment, and the cost of operation exclusive of that for air moving are normally large compared to the packing and air moving costs and therefore there is little reason to defend one of the 3 packing styles over the other two.

2. However, unless the packing material is very high in price or the electrical power rate is very low, of the types tested, the packing having the lower value for M would appear to be the least expensive based on a capitalized cost method of analysis.

CONCLUSIONS

1. The value of the transfer coefficient per unit volume "U" was found to be essentially the same for all heights of a given packing.

2. The value of the transfer coefficient was found to vary approximately as the 0.5 power of the mass water flow rate, L.

3. The value of the transfer coefficient was found to vary approximately as the 0.5 power of the mass air flow rate, G.

4. The transfer coefficient was found to vary approximately as the 0.5 power of the number of packing decks per ft of tower height, N.

5. The transfer coefficient was found to be the greatest per board foot for the packing having the highest value for the ratio of packing slat cross-section width to height, M.

6. The transfer coefficient was found to be the greatest per sq ft of surface area for the packing having the highest value for the ratio of packing slat cross section width to height, M.

7. The pressure drop varied approximately directly with the packing height.

8. The pressure drop varied approximately as the square of the mass air flow rate, G.

9. The pressure drop varied approximately as a linear function of the mass water flow rate, L.

10. The pressure drop attributed to air flow varied approximately directly with the number of packing decks per ft of tower height, N, while the pressure drop attributed to water flow varied approximately as the 1/3 power of N.

11. The pressure drop attributed to air flow was greatest for the high value of the ratio of packing slat cross-section width to height, M.

12. The pressure drop attributed to water flow changed little with M but showed a minimum at $M = 1.5$.

13. The pressure drop per board foot was greatest for the high value of M.

14. The pressure drop per unit value of transfer coefficient was greatest for the high value of M.

SYMBOLS

G = mass air flow rate, pounds per (hour) (square foot of ground area).
h' = enthalpy of saturated air at water temperature, Btu per pound dry air.
h_a = enthalpy of main air stream, Btu per pound dry air.
L = mass water flow rate, pound per (hour) (square foot of ground area).
M = ratio of packing slat cross-section width to height.
N = number of packing decks per foot of tower height.
P = pressure drop per foot of height, inches of water. (Is designated P' when obtained from Equation 4.)
P_G = pressure drop per foot of height attributed to air flow.
P_L = pressure drop per foot of height attributed to water flow.
R = coefficient in the pressure drop equation.
t = air temperature. Fahrenheit degrees.
T = water temperature, Fahrenheit degrees.
T_1 = water-on temperature, Fahrenheit degrees.
T_2 = water-off temperature, Fahrenheit degrees.
U = volume coefficient of mass transfer, pounds per (hour) (cubic foot) (unit vapor content potential) (pound per pound) (Is designated U' when obtained from Equation 3.)
V = cubic feet of total equivalent tower volume, cubic feet per square foot of ground area; or total height of equivalent tower, feet.
Z = packing height, feet.

DISCUSSION

J. G. DE FLON*, Whittier, Calif., (WRITTEN): I feel that Professor Smith is to be complimented for this fine paper in fundamental research. Once again it has been the academic world which is leading the way for industry in basic research. Professor Smith has shown a great deal of insight into cooling tower problems by carefully focusing attention on certain critical variables. By keeping the area of research clearly defined, he has been able to isolate those factors which could easily have been overlooked in a morass of complexities.

Professor Smith's contribution has served to crystallize in my own mind certain problems and conclusions which I have outlined below.

The overall performance of the packings tested appear to be somewhat higher than similar packings reported in the literature, and similar packings run in our own small test unit. This may be due to the manner in which corrections for wall water and end effects were applied. We found that the end effect corrections had to be applied in series with the overall performance, while the corrections for the effect of the wall water had to be more or less applied in parallel. To measure the end effects, the distributing system was placed just above the level of the bottom deck and the performance with this configuration plotted as UV/L versus L/G. This value was then subtracted from the overall performance of the particular packing under consideration. To determine the effects of the wall water, varying amounts of water were distributed over the walls

* Manager of Engineering, Fluor Products Company.

of the empty tower from which we determined the value of U for varying air rates. This value was then subtracted from the final data corrected for end effects when plotted as U *versus* G with the varying water rate as the paramater.

I feel that fundamental equation in which Professor Smith demonstrates that the mass transfer coefficient increases as the 0.5 power of the water rate, the gas rate, and the deck spacing, and decreases as an 0.2 power of the width to depth ratio of the slat is an important contribution. I do not feel, however, that enough work was done to accurately determine the coefficient for the width to depth ratio. If this exponent varied as the -0.2 power as he has indicated, one can show that one row of decks constructed of ⅜ in. x 2 in. slats horizontally spaced 1⅛ in. on center is more effective than 2 rows of ⅜ in. x 1 in. slats with the same horizontal spacing. To me this appears to be obviously untrue. A little more work on this factor would more accurately determine the exponential function to be applied to this coefficient.

Professor Smith has indicated that initially some difficulty was experienced in measuring the static pressure drop across the packing and that this was finally solved by some special instrumentation. We have never been entirely satisfied with our own pressure drop metering and would be interested in a description of the metering device finally employed.

I am hopeful that Professor Smith will continue his excellent work in the field of cooling tower research and that he will direct the interests of his students to this area. There is such a need for accurate scientific data in this industry that we cannot emphasize too strongly the value placed on this contribution.

KARL E. JOHNSON, Kansas City, Mo., (WRITTEN): This paper represents another contribution to the rather sparse literature on the performance and pressure drop characteristics of redwood packing arrangements similar to those in commercial use in counterflow cooling towers. The paper shows rather clearly that the performance of the cooling tower cannot be measured in terms of the surface area of the packing used; likewise, the projected (splash) area of the packing is not, in itself, an indication of the performance capability of the packing.

The mass transfer coefficient per board foot of packing material for the packings tested clearly indicates that the packing consisting of vertical slats is wasteful of wood in comparison to a packing made up of slats mounted horizontally. The main purpose of the packing materials commonly used in counterflow cooling towers is to break the water into finer drops in order to present a greater surface area to the air. For this purpose, a packing layer using slats of sufficient thickness to be self-supporting is all that is required. Any additional slat height merely adds a small amount of wetted surface that is essentially ineffectual.

The packings reported in the paper were tested at substantially closer vertical spacings than those generally used in industrial counterflow cooling towers. For that reason, the conclusions reached concerning the relative merits of the packings tested should not be applied to industrial tower packings without additional investigation. The first type of packing reported in the paper, consisting of ⅜ in. x 1 in. slats placed on edge on 1⅛ in. horizontal spacing was, as the author points out, used commercially for some years; however, this packing was used on 15 in. vertical centers in commercial towers rather than on 3, 6, or 12 in. vertical spacing, as reported in the paper. Performance and pressure drop data on this packing on 15 in. vertical spacing, together with performance and pressure drop characteristics of a number of other types of redwood packing arrangements, were reported in the Kelly-Swenson paper published in *Chemical Engineering Progress Magazine* in the July, 1956 issue. The data reported in that paper were obtained in a 6 ft x 6 ft counterflow test cell and the results were confirmed in tests on commercial size field cooling towers. Included in the report was a flat slat deck consisting of ⅜ in. x 2 in. slats mounted horizontally on 8 in. horizontal centers, with packing layers at 9 in. vertical centers. A comparison of these 2 decks, similar to that made in the present paper, showed that the vertical slat deck had only about

an 18 percent advantage over the flat slat deck in terms of pressure drop per unit value of transfer coefficient; however, despite the disparity in vertical spacing, the vertical slat deck had more than 30 percent additional redwood requirement. A thorough analysis of the total evaluated cost of cooling towers using these 2 decks indicates quite clearly that the flat slat deck described has a small advantage over the vertical slat deck on a capitalized cost basis of analysis. The present paper indicates that unless the packing material is very high in price, the vertical slat deck should be the least expensive on an evaluated basis. The fact of the matter is, in redwood cooling tower packings constructed of lath-like materials, the packing material is probably the most expensive wood in a cooling tower on a board foot basis. This fact probably explains why the vertical slat packing, which is wasteful of wood, is no longer in general use in industrial cooling towers. We might mention that the Kelly-Swenson paper referred to, in which the distribution system performance w as separated from the packed section performance, confirms reasonably well the present author's finding that the performance varies approximately as the one-half power of L, G, and the number of packing layers per foot of height. The Kelly-Swenson paper does not, however, support the finding that the performance is an inverse function of the ratio of slat width to slat height. Two of the decks reported were identical except for the ratio of slat width to slat height, and the performance and pressure drop characteristics showed no significant difference.

D. R. BAKER*, Kansas City, Mo.: First, I would like to make one comment on the system that ASHAE uses in publishing papers prior to the session. It gives all a chance to see the paper and come prepared to talk about it. From the journalistic standpoint maybe we lose the scoop, but I do not think this applies to a technical paper. I think the more revolutionary the paper is, the more interest there is in the paper. I would like to commend the step taken in doing this, and I think other societies should do the same thing.

As to the paper itself, I think Professor Smith is to be complimented. It is a wonderfully well written paper, well organized. Mr. De Flon made a point on pressure drop. It is an interesting point. It is one we have had trouble with. I was going to ask for information on this.

I was also going to ask how the author distributed the water. That is one of the most important factors in a cooling tower—to get equal water distribution throughout the test to get proper correlations.

A cooling tower is a most interesting piece of equipment. Towers follow the laws of nature but sometimes it is a little hard to follow the paths that nature follows. In this connection I am wondering if the paper didn't oversimplify and introduce approximations which probably make the paper more readable, but those in the industry are more concerned with the deviations than with the absolute consistency of the data. We like to see the deviations in there.

AUTHOR'S CLOSURE: I want to thank Mr. De Flon, Mr. Johnson, and Mr. Baker for their comments. Since Mr. De Flon's and Mr. Baker's remarks were of similar nature, directed toward the technical aspects of the tests themselves and the analysis, I shall first enlarge on the testing and correlation techniques.

A great deal of difficulty was experienced initially with water distribution, wall water, pressure drop measurements, and the evaluation of end effects. After many changes leading to improvement but not to complete satisfaction, the decision was made to continue with the packing evaluation on a comparative basis.

The water distribution system evolved from a nozzle system with much fine spray to a flume system with many small streams falling a sufficient distance before striking a splash surface above the packing itself to become broken into small droplets. This should not be construed to mean that flume distribution is superior to spray distribu-

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* The Marley Company.

tion in a commercial installation. We were not interested in the maximum transfer from the tower; we were interested only in spreading the water as uniformly as possible over the cross section. The first measurements of wall water revealed that approximately 50 percent of the total water flow was on the walls. The changes in the distribution system and the installation of baffles to scrub the water off of the walls reduced this to such a value that its affect was given no further consideration in the evaluations.

The first pressure drop values were obtained simply by measurement of the static pressure in the large plenum located below the transfer sections with and without packing in these sections. Because the pressure drops through the packing were small compared to the static pressures, the determinations were not entirely satisfactory. The pressure loss values reported in this paper were obtained by direct measurement of the difference between the total pressure in a section obtained by a large total pressure tube directed downward into the air stream, and the static pressure in the plenum chamber at a location having negligible air velocity. The pressure measurements are now being taken, however, by means of large piezometer rings completely surrounding the test section.

The end effects were accounted for by the extrapolation of the results obtained from the testing of the various packing heights. These end effects were expressed in terms of an equivalent height of the particular packing being tested. For example, note in test number 1026 where 3 ft of packing was used that the equivalent packing height was determined to be 2.11 ft.

In addition to eliminating or properly accounting for the foregoing items and similar ones, it was also necessary to reduce the overall problem to one with a sufficiently small number of variables to allow for meaningful correlation. Certain of the factors held constant during the tests reported here are now being investigated, and the remainder will be eventually. This *oversimplification* was considered as necessary in order to establish the effect of the variables under consideration. The data obtained were treated in an objective manner in the development of the reported relations and no attempt at rationalization was made until the reported relations were developed. Briefly the transfer correlation consisted of a time consuming study of the 9 groups of graphs similar to Fig. 3, the development from these of 3 additional groups with end effects accounted for, the development in turn of still another group of graphs and eventually one graph for which the equation was developed. Fig. 4 shows the amount of deviation of the measured values from the values calculated from the derived equation and therefore is an indication of the degree of approximation involved. The 4 points showing the greatest deviation were discovered to be in error and consequently I have asked that they be removed from the plate used in the printing of The TRANSACTIONS. It might add value to this information to know that points marked with circles were obtained from tests of packing consisting of slats having an M of 0.36; the crosses had to do with M of 1.00; and the exes (x) were for M of 2.78. The group of 3 points at (101, 109), (102, 117), and (103, 127) were obtained from tests of the flat packing ($M = 2.78$) using decks spaced on 12-in. centers, L/G ratios of approximately 0.86, and for 3, 6, and 9 ft respectively of packing heights.

One of the most pertinent problems in engineering has to do with the differences existing between laboratory and calculated values on the one hand, and results found in commercial installations on the other hand. Perhaps this could be stated in a more truthful and positive fashion by saying that one of the principal activities of an engineer is to translate proven scientific facts into useful consumer commodities. Needless to say, he must cope with situations and variables not even dreamed of by the scientist in his laboratory. The group responsible for the Cooling Tower Program at Texas A & M have not only had experience with precision laboratory types of determinations, with cooling towers as large as 8 by 8 ft designed specifically for experimental purposes, but also with field testing of towers ranging from those used on small domestic air conditioning installations to large units for refinery operation.

To reiterate, the results presented here are comparative but are based on measurements that can and have been duplicated. The present program includes various shapes of packing slats made from various materials and is pointed toward the development of a rational design procedure. We do not believe it possible at the moment to definitely state that one packing configuration is superior to all others. We could, having sufficient information including knowledge of the often nebulous economic aspects and the generally unpredictable social and political aspects of the problem, make recommendations as to the preferable choice from among existing configurations. It is hoped that eventually we will be able to *design* the optimum configuration for any given situation.

Exposition—W. G. Pennington, *Co-Chairman*; H. F. Abendroth, R. T. Flickinger, J. J. Friedler, Jr., John Kastholm, Jr., L. P. Lang, J. W. Muessig, L. E. Schulein, J. T. Wells.

ENTERTAINMENT: J. S. Kearney, *Chairman*

Orchestra, Acts, etc.—H. E. Anderson, *Co-Chairman*; W. J. Abraham, J. J. Floreth, C. O. Pratt, W. A. Stahl.

Ladies Activities—Mrs. H. G. Gragg, *Co-Chairman*; Mmes. H. E. Anderson, M. J. Bamond, M. W. Bishop, C. M. Burnam, Jr., H. J. Couch, G. G. Freyder, E. P. Heckel, Jr., J. S. Kearney, Herbert Kreisman, W. A. Kuechenberg, P. J. Marschall, L. O. Paul, W. G. Pennington, R. W. Roose, R. J. Salinger, J. C. Scott, H. C. Stevens, L. H. Streb, E. R. Teske, G. V. Zintel.

HOSPITALITY: Herbert Kreisman, *Chairman*

Reception—M. W. Bishop, *Co-Chairman*; R. D. Anderson, A. M. Bladen, G. W. Bornquist, B. L. Casey, H. G. Chapin, Melvin Fakter, J. M. Frank, G. D. Haberer, Louis Hatz, B. H. Jennings, S. R. Lewis, F. W. Luft, Jr., E. N. Patek, S. A. Pope, William Shelley, R. W. Stewart, W. T. Walters.

Registration—R. J. Salinger, *Co-Chairman*; A. B. Allen, M. B. Chalfen, M. E. Coobs, T. H. Cloke, Jr., O. P. Crose, R. N. Demir, G. R. Healy, G. A. Mansinger, F. W. Riederer.

Inspection Trips—J. C. Scott, *Co-Chairman*; N. E. Bueter, S. H. Rich.

Information—E. R. Teske, *Co-Chairman*; J. B. Hill, S. M. Miner, J. J. Phillipi, Henry Seetch, Jr.

Transportation—H. C. Stevens, *Co-Chairman*; S. O. Johnson, J. H. Lazar.

Finance—E. P. Heckel, Jr., *Chairman*; C. W. Blodgett, K. G. Carlson, E. I. Kadlec, Jr., S. C. Tarnoff, C. M. Vreuls.

SESSIONS: L. O. Paul, *Chairman*

Arthur Abrahamson, M. K. Arenberg, M. M. Hattis, C. P. Iseminger, B. A. Krulewitch, H. T. Kucera, W. S. Maxim, J. G. Pecis, D. K. Ridley, J. E. Sorensen, C. E. Tomlin, W. W. Treichler, Jr., A. R. Wallace.

PUBLICITY: R. W. Roose, *Chairman*

L. A. Erickson, E. H. Gage, J. H. Gazlay, T. V. Johnson, W. L. Lowry.

FACILITIES: H. J. Couch, *Chairman*

C. B. Bendix, A. C. De Smet, Edmund Gelfand, E. H. Gustafson, E. M. Kralovec, Jr., L. P. Lebow, Eugene Rovin, A. R. Thompson, Jr., E. J. Williams.

1591

No. 1591

STUDY OF THREE REDWOOD COOLING TOWER PACKINGS

By WESLEY W. SMITH*, COLLEGE STATION, TEX.

THREE types of redwood cooling tower packings were subjected to experimentation in efforts toward 3 objectives: (1) to obtain information of a fundamental nature on mass transfer and on resistance to air flow in various cooling tower packings; (2) to determine why individual cooling tower manufacturers have individual preferences as to packing types; and (3) to point the way toward improvement in packings.

The experimentation was accomplished in a 2- by 2-ft counterflow section which can handle packing arrangements up to 9 ft in height. It is an indoor installation specifically designed for evaluating[1] packing and other cooling tower components (see Fig. 1).

Several hundred tests performed with this equipment over a period of more than a year are the foundation for the following presentation.

DESCRIPTION OF PACKING AND DETAILS OF PROBLEM

In an effort to obtain as much information as possible on many of the variables known to affect cooling tower performance without creating an impasse of complexity, the testing was planned for 3 readily prepared and related packing styles each to be used with 3 vertical spacings and 3 thicknesses. Water flow rates ranging from approximately 1½ to 4½ gpm per sq ft and air velocities of from 275 to 550 fpm were considered as reasonable and easily obtained. Manual control of the water-on temperature at approximately 105 F was considered feasible but control of the wet-bulb and water-off temperatures too difficult and time consuming with the present equipment.

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* Associate Research Engineer, Texas Engineering Experiment Station, Texas A. & M. College System.

[1] A Test Section for Evaluating Cooling Tower Components, by A. B. Alter (M.S. Thesis, A. & M. College of Texas, College Station, Tex., May 1953).

Presented at the 63rd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Chicago, February 1957.

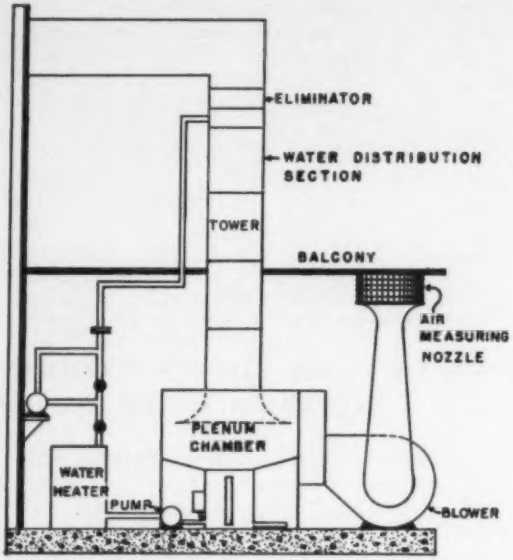


FIG. 1—FACILITY FOR 2- BY 2-FT COUNTERFLOW TESTS

TABLE 1—DETAILS OF PACKINGS AND PERFORMANCE DATA

ITEM NO.		TOWER PACKINGS TESTED		
		FIRST TYPE	SECOND TYPE	THIRD TYPE
1	Ratio of packing slat cross-section—width/height, (M)	0.36	1.00	2.78
2	Slat width, inches	0.36	0.59	1.00
3	Slat height, inches	1.00	0.59	0.36
4	Slat cross-sectional area, sq in.	0.36	0.35	0.36
5	Number of slats per deck	21.	13.	8.
6	Projected area, (sq ft), per deck per sq ft of tower ground area	0.298	0.306	0.316
7	Board ft of material per deck per sq ft of tower ground area	0.298	0.181	0.113
8	Surface area, (sq ft), per deck per sq ft of tower ground area	2.26	1.21	0.86
9	Volume coefficient of mass transfer, U[b], (for $N = 1$; $L = 1250$; and $G = 1250$)	130.4	106.2	86.6
10	U per board ft (Item 9 ÷ Item 7)	438.	587.	765.
11	U per sq ft proj. area (Item 9 ÷ Item 6)	438.	347.	274.
12	U per sq ft surf. area (Item 9 ÷ Item 8)	57.7	87.8	100.6
13	P, (pressure drop, inches water, per ft of height (for $N = 1$; $L = 1250$; $G = 1250$)	0.00681	0.00690	0.00954
14	P, per board foot (Item 13 ÷ Item 7)	0.0228	0.0381	0.0843
15	P/U (Item 13 ÷ Item 9)	0.522×10^{-4}	0.650×10^{-4}	1.102×10^{-4}

[b] The value of the coefficient U was obtained from Equation 3.

Fig. 2 and Table 1 show the details of the first packing tested. This 2-ft square packing unit made of $\frac{3}{8} \times 1$ in. nominal dimension redwood slats placed on edge is a standard packing used by 2 large manufacturers of cooling towers.

The second type (Table 1) was made from 13 square slats each having a cross-sectional area of 0.35 sq in. spaced to maintain essential constancy of projected area.

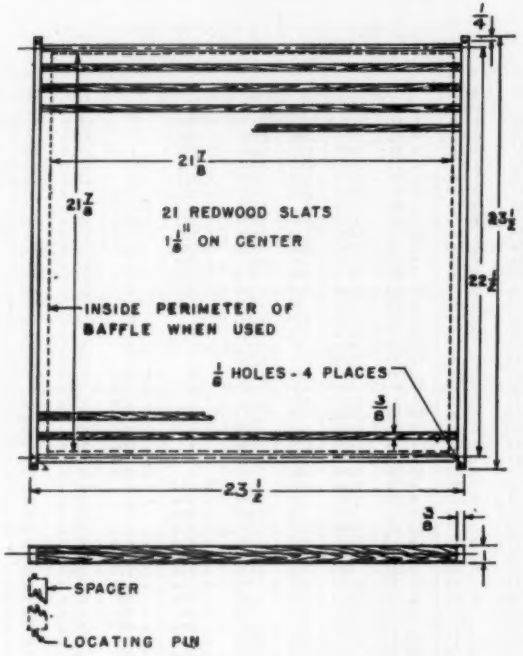


FIG. 2—DETAILS OF FIRST PACKING TESTED

The third type of packing tested (Table 1) was constructed of 8 of the same slats but arranged flatwise and the smaller number spaced to give essentially the same projected area as in the first type.

The 2 side pieces were $\frac{3}{8} \times 1$ in. for all the packing units.

It should be noted that the slats used in all the packings have essentially the same cross-sectional area and that all packing units have essentially the same projected area. The number of slats and the amount of total surface area vary with the ratio, M, of slat width to height.

The mass transfer per cubic foot of tower volume per unit vapor content potential was taken as the criterion of performance and was obtained from the correlation of the results of numerical integration of the commonly used cooling tower equation:[2]

$$UV/L = \int_{T_2}^{T_1} \frac{dT}{h'' - h_a} \qquad \text{(1)}$$

TABLE 2—TYPICAL DATA OBTAINED IN TESTS

TEST NO.	L/G	UV/L	V	L	U^a	G	$M^{-0.2b}$	$N^{0.8}$	$L^{0.8}$	$G^{0.8}$	U'^c	N	Z
1026	1.50	0.48	5.11	2288	215	1526	1.227	1	47.8	39.1	195	1	3
1028	0.51	0.63	5.11	814	100	1596	1.227	1	28.5	40.9	119	1	3
1031	0.86	0.88	8.11	1388	151	1606	1.227	1	37.2	40.1	155	1	6
1034	0.52	1.52	11.11	839	115	1603	1.227	1	29.0	40.0	121	1	9
1036	1.43	1.00	11.11	2278	205	1597	1.227	1	47.7	40.0	199	1	9
078	0.87	0.69	4.49	1395	214	1607	1.227	1.414	37.4	40.1	221	2	3
980	1.41	0.92	7.49	2291	281	1625	1.227	1.414	47.9	40.3	284	2	6
982	0.54	1.51	7.49	878	177	1637	1.227	1.414	29.6	40.5	177	2	6
984	0.85	1.69	10.49	1391	224	1638	1.227	1.414	37.3	40.5	223	2	9
1019	0.88	0.89	4.06	1394	306	1391	1.227	2	37.4	39.9	311	4	3
1014	1.41	1.26	7.06	2273	405	1611	1.227	2	47.7	40.1	398	4	6
1016	0.55	2.10	7.06	874	260	1590	1.277	2	29.6	39.9	246	4	6
1002	0.86	2.17	10.06	1389	300	1606	1.227	2	37.3	40.1	312	4	9
1073	0.54	0.62	5.59	861	96	1587	1	1	29.3	39.8	99	1	3
1075	1.43	0.44	5.59	2278	179	1596	1	1	47.7	40.0	162	1	3
1077	0.87	0.78	8.59	1390	126	1601	1	1	37.3	40.0	127	1	6
1080	0.55	1.27	11.59	883	97	1604	1	1	29.7	40.0	101	1	9
1082	1.41	0.90	11.59	2260	175	1599	1	1	47.5	40.0	162	1	9
1085	0.87	0.63	4.83	1394	182	1606	1	1.414	37.3	40.1	180	2	3
1087	0.52	1.20	7.83	839	129	1605	1	1.414	29.0	40.1	140	2	6
1089	1.44	0.82	7.83	2281	239	1589	1	1.414	47.8	39.9	229	2	6
1092	0.85	1.38	10.83	1394	178	1633	1	1.414	37.3	40.4	181	2	9
1096	0.86	0.82	4.29	1389	265	1608	1	2.000	37.3	40.1	254	4	3
1098	1.42	1.01	7.29	2280	316	1609	1	2.000	47.8	40.1	326	4	6
1100	0.54	1.74	7.29	873	208	1606	1	2.000	29.5	40.1	203	4	6
1103	0.87	1.81	10.29	1386	244	1595	1	2.000	37.2	39.9	252	4	9
1047	0.84	0.50	6.18	1350	109	1606	0.814	1	36.7	40.1	101	1	3
1042	1.43	0.64	9.18	2283	159	1594	0.814	1	47.8	39.9	132	1	6
1044	0.55	0.95	9.18	881	91	1592	0.814	1	29.7	39.9	82	1	6
1039	0.88	1.11	12.18	1400	127	1597	0.814	1	37.4	40.0	103	1	9
1051	0.87	0.57	5.25	1385	150	1596	0.814	1.414	37.2	40.0	146	2	3
1054	1.40	0.76	8.25	2281	210	1627	0.814	1.414	47.8	40.3	188	2	6
1056	0.55	1.06	8.25	883	114	1610	0.814	1.414	29.7	40.1	116	2	6
1059	0.87	1.23	11.25	1388	152	1592	0.814	1.414	37.2	39.9	145	2	9
1062	1.44	0.55	4.59	2283	273	1582	0.814	2.00	47.8	39.8	263	4	3
1064	0.53	0.87	4.59	839	159	1588	0.814	2.00	29.0	39.8	160	4	3
1066	0.88	1.17	7.59	1389	214	1582	0.814	2.00	37.3	39.8	205	4	6
1069	1.41	1.15	10.59	2288	249	1619	0.814	2.00	47.8	40.2	266	4	9
1071	0.53	1.68	10.59	839	133	1586	0.814	2.00	29.0	39.8	160	4	9

[a] Obtained from test and integral.
[b] When $M^{-0.2}$ is 1, $M = 1$; when $M^{-0.2}$ is 0.814, $M = 2.78$; when $M^{-0.2}$ is 1.227, $M = 0.36$.
[c] Obtained from $U' = 0.085\ M^{-0.2}\ N^{0.8}\ L^{0.8}\ G^{0.8}$ (Equation 3).

Measuring of the makeup water was eliminated after the initial tests as values required for the evaluation of Equation 1 could be obtained without it.

The initial pressure drop measurements were obtained by the subtraction of the positive pressure in the plenum chamber with an unpacked tower from that with

[2] HEATING, VENTILATING, AIR-CONDITIONING GUIDE, 1952, p. 760.

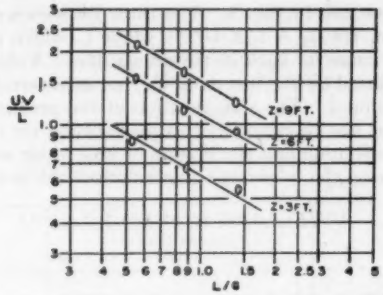


FIG. 3—TOWER TRANSFER CHARACTERISTICS

the subject packing. As the resistance to air flow through the eliminator and other parts of the tower was large compared to the resistance in the packing, the initial measurements were not entirely satisfactory. Therefore a special total pressure pickup device was installed resulting in more consistent results.

MASS TRANSFER COEFFICIENT

To illustrate the general method used in the following development and to give some idea of the degree of approximation involved, typical test values of the dimensionless group, UV/L, with the corresponding values of the dimensionless

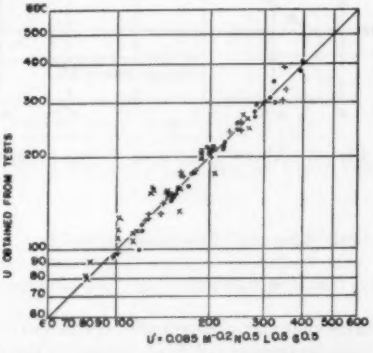


FIG. 4—DEVIATION OF MASS TRANSFER TEST DATA FROM VALUES GIVEN BY THE DERIVED EQUATION

group, L/G, are presented in Fig. 3. The three lines shown are represented by the general equation, $UV/L = C(L/G)^{-0.5}$, where C, which is also dimensionless, has a definite fixed value for each of the drawn lines. Values for C are entirely empirical, being dictated by the test data, and are an intermediate step in the development of Equation 2. The simplification of the problem resulting from the approximation might not be considered as justification for its use if it were not that even water distribution and elimination of wall water were not attained. It was possible to develop the following equations from the test data.

$$UV/L = 0.085\ M^{-0.2}\ N^{0.5}\ V(L/G)^{-0.5} \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

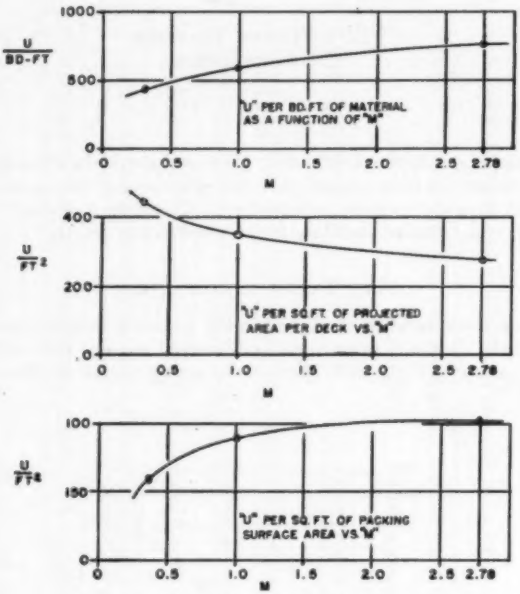


FIG. 5—MASS TRANSFER COMPARISONS

Eliminating V from both sides and transposing L yields

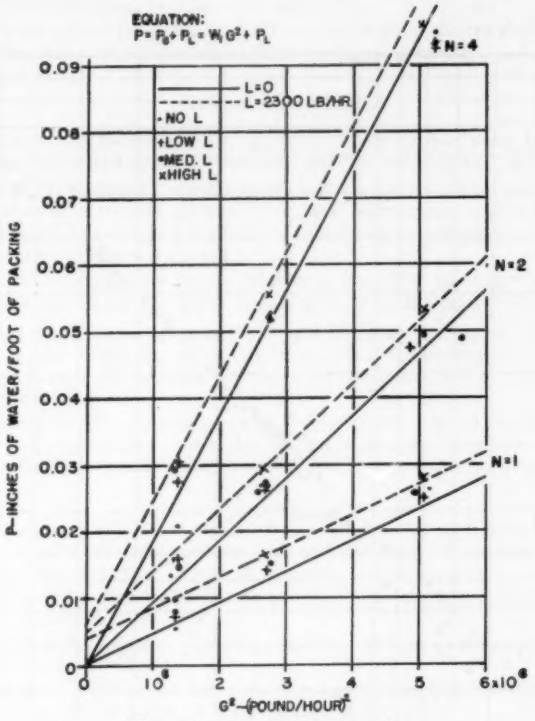
$$U = 0.085\ M^{-0.2}\ N^{0.5}\ L^{0.5}\ G^{0.5} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

This Equation 3 states that the mass transfer coefficient, U, increases as the 0.5 power of L, G and N, but decreases with the 0.2 power of the ratio of width to depth of slat cross-section.

All values of the coefficient U obtained from test and Equation 1 are shown in Fig. 4 with the corresponding values obtained from Equation 3. The agreement

is considered good in view of the complexity of the problem and the known deficiencies of control.

For the conditions $L = G = 1250$ lb per hr and $N = 1$, the value for U per board foot of packing material is presented in Fig. 5 as a function of M as is also U per



(Figure shows pressure drop per foot of packing for the tests when $M = 2.78$)

FIG. 6—TYPICAL VALUES OF PRESSURE DROP OBTAINED BY TESTS

sq ft of projected area per deck and U per sq ft of surface area. Table 1 contains the values just mentioned and also the important physical dimensions on the 3 packing styles.

PRESSURE DROP

Resistance proved to be essentially linear with packing height, and therefore few measurements were taken except for the maximum number of sections. Calibra-

tion pressure drop values obtained for the facility with no sections were subtracted from the 3 section manometer readings before calculating P. Since the resistance with no water flow appeared to be very nearly proportional to the flow indicating manometer reading, it was plotted as a function of G^2. Fig. 6 shows typical values of pressure drop obtained by test. The additional resistance caused by the water

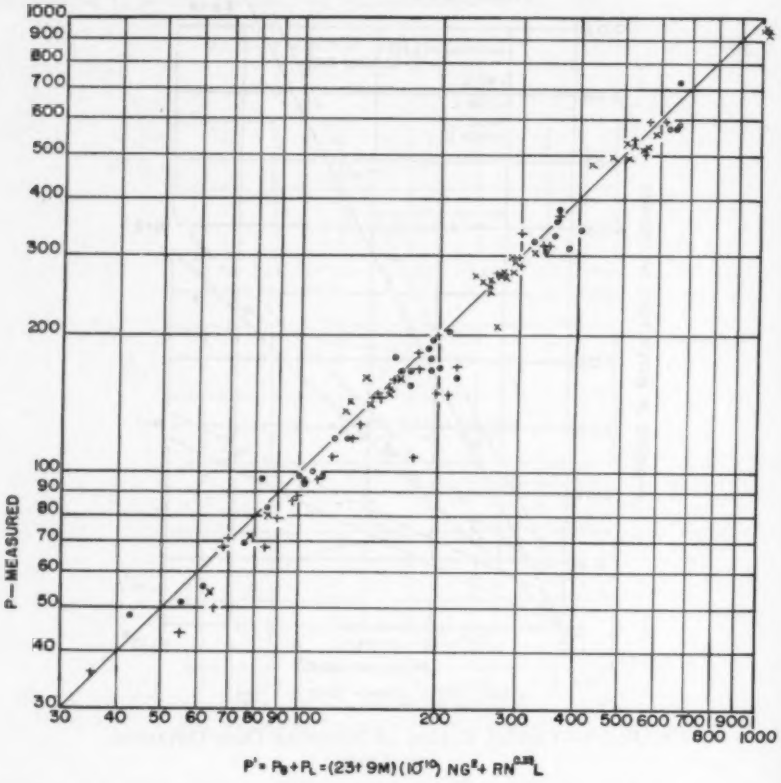


FIG. 7—DEVIATION OF PRESSURE LOSS TEST DATA FROM VALUES GIVEN BY THE DERIVED EQUATION

is represented as a constant value for any packing configuration and air flow and for simplicity is considered linear with water flow. The solid lines represent the no-water resistance, the dotted lines the resistance with 2300 lb of water per hr per sq ft of tower cross section.

An analysis of these data led to the development of an equation for the resistance to air flow as a function of the packing style, spacing amount, and rates:

$$P = P_G + P_L = (23 + 9M)\ (10^{-10})\ NG^2 + RN^{0.33}L \quad . \quad . \quad . \quad . \quad (4)$$

Note that R takes on the value of 2.17×10^{-6}, 1.52×10^{-6}, and 1.65×10^{-6} respectively for M values of 0.36, 1.00, and 2.78.

All measured values for pressure drop are shown as ordinates in Fig. 7 with the corresponding values calculated from Equation 4 as abscissas.

COMPARATIVE COSTS

As in so many engineering problems, the final comparison must be made on an economic basis in order to evaluate properly both the investment costs and operating costs. However, since each cooling tower installation has its own set of economic variables—cost of materials, cost of land, local restrictions, cost of labor, cost of power, water treatment and availability, conditions of operation, and others, no formal treatment will be attempted.

Nevertheless, comparative cost calculations based on what are believed to be representative values for the many variables lead to the following statements:

1. The cost of the tower structure itself, the accessory equipment, and the cost of operation exclusive of that for air moving are normally large compared to the packing and air moving costs and therefore there is little reason to defend one of the 3 packing styles over the other two.

2. However, unless the packing material is very high in price or the electrical power rate is very low, of the types tested, the packing having the lower value for M would appear to be the least expensive based on a capitalized cost method of analysis.

CONCLUSIONS

1. The value of the transfer coefficient per unit volume "U" was found to be essentially the same for all heights of a given packing.

2. The value of the transfer coefficient was found to vary approximately as the 0.5 power of the mass water flow rate, L.

3. The value of the transfer coefficient was found to vary approximately as the 0.5 power of the mass air flow rate, G.

4. The transfer coefficient was found to vary approximately as the 0.5 power of the number of packing decks per ft of tower height, N.

5. The transfer coefficient was found to be the greatest per board foot for the packing having the highest value for the ratio of packing slat cross-section width to height, M.

6. The transfer coefficient was found to be the greatest per sq ft of surface area for the packing having the highest value for the ratio of packing slat cross section width to height, M.

7. The pressure drop varied approximately directly with the packing height.

8. The pressure drop varied approximately as the square of the mass air flow rate, G.

9. The pressure drop varied approximately as a linear function of the mass water flow rate, L.

10. The pressure drop attributed to air flow varied approximately directly with the number of packing decks per ft of tower height, N, while the pressure drop attributed to water flow varied approximately as the 1/3 power of N.

11. The pressure drop attributed to air flow was greatest for the high value of the ratio of packing slat cross-section width to height, M.

12. The pressure drop attributed to water flow changed little with M but showed a minimum at $M = 1.5$.

13. The pressure drop per board foot was greatest for the high value of M.

14. The pressure drop per unit value of transfer coefficient was greatest for the high value of M.

SYMBOLS

G = mass air flow rate, pounds per (hour) (square foot of ground area).
h'' = enthalpy of saturated air at water temperature, Btu per pound dry air.
h_a = enthalpy of main air stream, Btu per pound dry air.
L = mass water flow rate, pound per (hour) (square foot of ground area).
M = ratio of packing slat cross-section width to height.
N = number of packing decks per foot of tower height.
P = pressure drop per foot of height, inches of water. (Is designated P' when obtained from Equation 4.)
P_G = pressure drop per foot of height attributed to air flow.
P_L = pressure drop per foot of height attributed to water flow.
R = coefficient in the pressure drop equation.
t = air temperature. Fahrenheit degrees.
T = water temperature, Fahrenheit degrees.
T_1 = water-on temperature, Fahrenheit degrees.
T_2 = water-off temperature, Fahrenheit degrees.
U = volume coefficient of mass transfer, pounds per (hour) (cubic foot) (unit vapor content potential) (pound per pound) (Is designated U' when obtained from Equation 3.)
V = cubic feet of total equivalent tower volume, cubic feet per square foot of ground area; or total height of equivalent tower, feet.
Z = packing height, feet.

DISCUSSION

J. G. DE FLON*, Whittier, Calif., (WRITTEN): I feel that Professor Smith is to be complimented for this fine paper in fundamental research. Once again it has been the academic world which is leading the way for industry in basic research. Professor Smith has shown a great deal of insight into cooling tower problems by carefully focusing attention on certain critical variables. By keeping the area of research clearly defined, he has been able to isolate those factors which could easily have been overlooked in a morass of complexities.

Professor Smith's contribution has served to crystallize in my own mind certain problems and conclusions which I have outlined below.

The overall performance of the packings tested appear to be somewhat higher than similar packings reported in the literature, and similar packings run in our own small test unit. This may be due to the manner in which corrections for wall water and end effects were applied. We found that the end effect corrections had to be applied in series with the overall performance, while the corrections for the effect of the wall water had to be more or less applied in parallel. To measure the end effects, the distributing system was placed just above the level of the bottom deck and the performance with this configuration plotted as UV/L versus L/G. This value was then subtracted from the overall performance of the particular packing under consideration. To determine the effects of the wall water, varying amounts of water were distributed over the walls

* Manager of Engineering, Fluor Products Company.

of the empty tower from which we determined the value of U for varying air rates. This value was then subtracted from the final data corrected for end effects when plotted as U *versus* G with the varying water rate as the paramater.

I feel that fundamental equation in which Professor Smith demonstrates that the mass transfer coefficient increases as the 0.5 power of the water rate, the gas rate, and the deck spacing, and decreases as an 0.2 power of the width to depth ratio of the slat is an important contribution. I do not feel, however, that enough work was done to accurately determine the coefficient for the width to depth ratio. If this exponent varied as the -0.2 power as he has indicated, one can show that one row of decks constructed of ⅜ in. x 2 in. slats horizontally spaced 1⅛ in. on center is more effective than 2 rows of ⅜ in. x 1 in. slats with the same horizontal spacing. To me this appears to be obviously untrue. A little more work on this factor would more accurately determine the exponential function to be applied to this coefficient.

Professor Smith has indicated that initially some difficulty was experienced in measuring the static pressure drop across the packing and that this was finally solved by some special instrumentation. We have never been entirely satisfied with our own pressure drop metering and would be interested in a description of the metering device finally employed.

I am hopeful that Professor Smith will continue his excellent work in the field of cooling tower research and that he will direct the interests of his students to this area. There is such a need for accurate scientific data in this industry that we cannot emphasize too strongly the value placed on this contribution.

KARL E. JOHNSON, Kansas City, Mo., (WRITTEN): This paper represents another contribution to the rather sparse literature on the performance and pressure drop characteristics of redwood packing arrangements similar to those in commercial use in counterflow cooling towers. The paper shows rather clearly that the performance of the cooling tower cannot be measured in terms of the surface area of the packing used; likewise, the projected (splash) area of the packing is not, in itself, an indication of the performance capability of the packing.

The mass transfer coefficient per board foot of packing material for the packings tested clearly indicates that the packing consisting of vertical slats is wasteful of wood in comparison to a packing made up of slats mounted horizontally. The main purpose of the packing materials commonly used in counterflow cooling towers is to break the water into finer drops in order to present a greater surface area to the air. For this purpose, a packing layer using slats of sufficient thickness to be self-supporting is all that is required. Any additional slat height merely adds a small amount of wetted surface that is essentially ineffectual.

The packings reported in the paper were tested at substantially closer vertical spacings than those generally used in industrial counterflow cooling towers. For that reason, the conclusions reached concerning the relative merits of the packings tested should not be applied to industrial tower packings without additional investigation. The first type of packing reported in the paper, consisting of ⅜ in. x 1 in. slats placed on edge on 1⅛ in. horizontal spacing was, as the author points out, used commercially for some years; however, this packing was used on 15 in. vertical centers in commercial towers rather than on 3, 6, or 12 in. vertical spacing, as reported in the paper. Performance and pressure drop data on this packing on 15 in. vertical spacing, together with performance and pressure drop characteristics of a number of other types of redwood packing arrangements, were reported in the Kelly-Swenson paper published in *Chemical Engineering Progress Magazine* in the July, 1956 issue. The data reported in that paper were obtained in a 6 ft x 6 ft counterflow test cell and the results were confirmed in tests on commercial size field cooling towers. Included in the report was a flat slat deck consisting of ⅜ in. x 2 in. slats mounted horizontally on 8 in. horizontal centers, with packing layers at 9 in. vertical centers. A comparison of these 2 decks, similar to that made in the present paper, showed that the vertical slat deck had only about

an 18 percent advantage over the flat slat deck in terms of pressure drop per unit value of transfer coefficient; however, despite the disparity in vertical spacing, the vertical slat deck had more than 30 percent additional redwood requirement. A thorough analysis of the total evaluated cost of cooling towers using these 2 decks indicates quite clearly that the flat slat deck described has a small advantage over the vertical slat deck on a capitalized cost basis of analysis. The present paper indicates that unless the packing material is very high in price, the vertical slat deck should be the least expensive on an evaluated basis. The fact of the matter is, in redwood cooling tower packings constructed of lath-like materials, the packing material is probably the most expensive wood in a cooling tower on a board foot basis. This fact probably explains why the vertical slat packing, which is wasteful of wood, is no longer in general use in industrial cooling towers. We might mention that the Kelly-Swenson paper referred to, in which the distribution system performance w as separated from the packed section performance, confirms reasonably well the present author's finding that the performance varies approximately as the one-half power of L, G, and the number of packing layers per foot of height. The Kelly-Swenson paper does not, however, support the finding that the performance is an inverse function of the ratio of slat width to slat height. Two of the decks reported were identical except for the ratio of slat width to slat height, and the performance and pressure drop characteristics showed no significant difference.

D. R. BAKER*, Kansas City, Mo.: First, I would like to make one comment on the system that ASHAE uses in publishing papers prior to the session. It gives all a chance to see the paper and come prepared to talk about it. From the journalistic standpoint maybe we lose the scoop, but I do not think this applies to a technical paper. I think the more revolutionary the paper is, the more interest there is in the paper. I would like to commend the step taken in doing this, and I think other societies should do the same thing.

As to the paper itself, I think Professor Smith is to be complimented. It is a wonderfully well written paper, well organized. Mr. De Flon made a point on pressure drop. It is an interesting point. It is one we have had trouble with. I was going to ask for information on this.

I was also going to ask how the author distributed the water. That is one of the most important factors in a cooling tower—to get equal water distribution throughout the test to get proper correlations.

A cooling tower is a most interesting piece of equipment. Towers follow the laws of nature but sometimes it is a little hard to follow the paths that nature follows. In this connection I am wondering if the paper didn't oversimplify and introduce approximations which probably make the paper more readable, but those in the industry are more concerned with the deviations than with the absolute consistency of the data. We like to see the deviations in there.

AUTHOR'S CLOSURE: I want to thank Mr. De Flon, Mr. Johnson, and Mr. Baker for their comments. Since Mr. De Flon's and Mr. Baker's remarks were of similar nature, directed toward the technical aspects of the tests themselves and the analysis, I shall first enlarge on the testing and correlation techniques.

A great deal of difficulty was experienced initially with water distribution, wall water, pressure drop measurements, and the evaluation of end effects. After many changes leading to improvement but not to complete satisfaction, the decision was made to continue with the packing evaluation on a comparative basis.

The water distribution system evolved from a nozzle system with much fine spray to a flume system with many small streams falling a sufficient distance before striking a splash surface above the packing itself to become broken into small droplets. This should not be construed to mean that flume distribution is superior to spray distribu-

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* The Marley Company.

tion in a commercial installation. We were not interested in the maximum transfer from the tower; we were interested only in spreading the water as uniformly as possible over the cross section. The first measurements of wall water revealed that approximately 50 percent of the total water flow was on the walls. The changes in the distribution system and the installation of baffles to scrub the water off of the walls reduced this to such a value that its affect was given no further consideration in the evaluations.

The first pressure drop values were obtained simply by measurement of the static pressure in the large plenum located below the transfer sections with and without packing in these sections. Because the pressure drops through the packing were small compared to the static pressures, the determinations were not entirely satisfactory. The pressure loss values reported in this paper were obtained by direct measurement of the difference between the total pressure in a section obtained by a large total pressure tube directed downward into the air stream, and the static pressure in the plenum chamber at a location having negligible air velocity. The pressure measurements are now being taken, however, by means of large piezometer rings completely surrounding the test section.

The end effects were accounted for by the extrapolation of the results obtained from the testing of the various packing heights. These end effects were expressed in terms of an equivalent height of the particular packing being tested. For example, note in test number 1026 where 3 ft of packing was used that the equivalent packing height was determined to be 2.11 ft.

In addition to eliminating or properly accounting for the foregoing items and similar ones, it was also necessary to reduce the overall problem to one with a sufficiently small number of variables to allow for meaningful correlation. Certain of the factors held constant during the tests reported here are now being investigated, and the remainder will be eventually. This *oversimplification* was considered as necessary in order to establish the effect of the variables under consideration. The data obtained were treated in an objective manner in the development of the reported relations and no attempt at rationalization was made until the reported relations were developed. Briefly the transfer correlation consisted of a time consuming study of the 9 groups of graphs similar to Fig. 3, the development from these of 3 additional groups with end effects accounted for, the development in turn of still another group of graphs and eventually one graph for which the equation was developed. Fig. 4 shows the amount of deviation of the measured values from the values calculated from the derived equation and therefore is an indication of the degree of approximation involved. The 4 points showing the greatest deviation were discovered to be in error and consequently I have asked that they be removed from the plate used in the printing of The TRANSACTIONS. It might add value to this information to know that points marked with circles were obtained from tests of packing consisting of slats having an M of 0.36; the crosses had to do with M of 1.00; and the exes (x) were for M of 2.78. The group of 3 points at (101, 109), (102, 117), and (103, 127) were obtained from tests of the flat packing ($M = 2.78$) using decks spaced on 12-in. centers, L/G ratios of approximately 0.86, and for 3, 6, and 9 ft respectively of packing heights.

One of the most pertinent problems in engineering has to do with the differences existing between laboratory and calculated values on the one hand, and results found in commercial installations on the other hand. Perhaps this could be stated in a more truthful and positive fashion by saying that one of the principal activities of an engineer is to translate proven scientific facts into useful consumer commodities. Needless to say, he must cope with situations and variables not even dreamed of by the scientist in his laboratory. The group responsible for the Cooling Tower Program at Texas A & M have not only had experience with precision laboratory types of determinations, with cooling towers as large as 8 by 8 ft designed specifically for experimental purposes, but also with field testing of towers ranging from those used on small domestic air conditioning installations to large units for refinery operation.

To reiterate, the results presented here are comparative but are based on measurements that can and have been duplicated. The present program includes various shapes of packing slats made from various materials and is pointed toward the development of a rational design procedure. We do not believe it possible at the moment to definitely state that one packing configuration is superior to all others. We could, having sufficient information including knowledge of the often nebulous economic aspects and the generally unpredictable social and political aspects of the problem, make recommendations as to the preferable choice from among existing configurations. It is hoped that eventually we will be able to *design* the optimum configuration for any given situation.

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No. 1592

EARTH AS HEAT SOURCE AND SINK FOR HEAT PUMPS†

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THE ATMOSPHERE, surface and ground water, and the earth itself are 3 principal media which may serve as the heat source and sink of a heat pump system.

Increased interest in the heat pump as a year 'round air-conditioning system has demanded that quantitative information on each of these media be obtained in order that a designer may make an engineering evaluation of the advantages and disadvantages of each for a particular installation. Each of the media and combinations of them have been studied by other investigators, listed in the Bibliography in the Appendix, but no generalized design criteria were evolved. This paper presents the results from an extensive investigation of the earth at shallow depths as a heat source and sink for the heat pump and gives the buried coil design equation which was developed. Details of the design method based on this work were given previously.[1]

The operating characteristics of horizontally buried coils of ½-in., 1-in., and 2-in. diameter were observed and studied under controlled operating conditions. Data were obtained for both the cases where the earth acted as a heat source, and where it acted as a heat sink. Results were combined with data from laboratory tests of artificially prepared soil specimens and with field measurements of natural soil temperatures into a correlation from which the previously mentioned design equation and design procedure were developed.

EQUIPMENT AND APPARATUS

A plot plan of the field installation where the buried coil testing program was conducted is presented in Fig. 1.

Construction of the instrument house and installation of the first four coils, A, B, C and D, was begun in the spring of 1949. Each of the coils A, B, C and

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† The data presented here were obtained in experiments conducted for the Texas Electric Service Co., Ft. Worth, by the Texas Engineering Experiment Station cooperatively with the Texas A. & M. Research Foundation.

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[1] Exponent numerals refer to References.

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D was independent of the others and was buried horizontally 5 ft below the ground surface. Experimental work on these coils was initiated in November, 1949.

When it became evident from data of the first 5 tests of these original 4 coils that the temperature change of the circulating fluid was too small to be measured accurately except at low rates of fluid flow, each pair of coils of like diameter was connected with a 10-ft long crossline. The coil length traveled by the fluid was thereby increased to a nominal 160 ft as shown in Fig. 1. All tests after the first 5 employed the *U*-shaped arrangement.

In the fall of 1951 the ½-in. diameter coils, designated E and F in Fig. 1, were added and were buried at the same depth as the other coils.

A schematic flow diagram of the field installation is shown in Fig. 2. Addi-

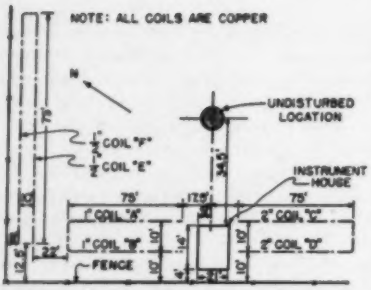

FIG. 1—PLOT PLAN OF FIELD INSTALLATION

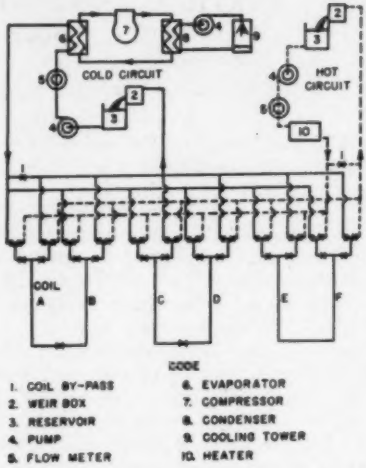

FIG. 2—SCHEMATIC FLOW DIAGRAM OF FIELD INSTALLATION

tional details of the equipment and apparatus employed are given in the Appendix under the heading *Details of Equipment and Apparatus*.

TEST PROCEDURE

Procedures followed in starting a test, in extending it over the desired period of time, and in ending it were standardized where possible for the bulk of the buried coil testing program. Early tests in which the fluid was circulated through only one coil 75 ft in length provided the basis for establishing many of the standard operating procedures. Others were modified or became standard as additional experience accumulated.

The general procedure followed in beginning a test run and in placing the buried coil into operation was as follows.

1. Selection of the Coil Pair to be Tested:—Selection of a coil pair was governed principally by the temperature pattern in the soil around the coil pair in question. Suffi-

cient time was allowed between 2 series of tests on the same pair to permit the soil temperatures to approach their natural undisturbed values. The coil testing plan adopted in which a series of 3 test runs were completed on each of the 3 coil pairs in succession gave ample time for soil temperature recovery between series on the same coil.

2. Selection of the Coil Inlet Fluid Temperature to be Maintained Constant Throughout the Test Run:—For coil condenser type runs, the coil inlet fluid temperature was selected between the limits of 90 to 130 F while for coil evaporator runs the limits were from 20 to 45 F. In each type of test, the range included coil temperatures anticipated in the operation of an actual buried coil heat pump installation used for comfort space conditioning. Selection of either a condenser or an evaporator test was governed, as would be the case in an installation used solely for comfort space conditioning, by the seasonal variation of the outside air temperature. That is, circulating fluid temperatures in the winter months were selected in the lower temperature range to simulate coil evaporator operation while those selected for the summer months were in the upper temperature range to simulate coil condenser operation.

3. Selection of an Average Reynold's Number and Computation of the Corresponding Rate of Fluid Flow to be Maintained Constant Throughout the Test Run:—Normally, a series of 3 tests in succession were made on the same coil size at the same coil inlet temperature. Only the Reynold's number was varied between tests in a series. Values of 1000, 2500, and 5000, respectively, were approximated depending upon the ability of the controlling devices to hold a constant fluid temperature and rate of flow.

4. Selection of Either Continuous or Intermittent Coil Operation:—Fluid was circulated with no interruption through the buried coils for the duration of a continuous coil operation test. In those tests in which intermittent operation was employed, 3 different cycles were devised lasting 2 hours each. The 3 intermittent cycles were ¼ cycle on, designated Case I; ½ cycle on, designated Case II; and ¾ cycle on, designated Case III. Two other cases, IV and V, combined these 3 basic intermittent cycles. Case IV cycle was composed of Cases I, II, and III in succession while Case V cycle was made up of Cases I, III, and II in that order. Each of the latter cases, as tests proved, was equivalent to ½ cycle on, designated Case II, and is reported as such.

The fraction of the cycle *on* refers to the initial part of the 2-hr period in which the fluid was circulated through the coil. During the remaining part of the cycle, the fluid flowed through only the coil bypass line and the particular common circuit involved in order to maintain flow rate and temperature for the next cycle. Switching of the fluid flow from coil to coil bypass line and back to the coil was accomplished by the simultaneous opposite action of 2 solenoid valves whose operation was controlled by an electric automatic on-off timer.

5. Regulation of Valves in the Fluid Circuit to Route the Fluid Through the Desired Flow Path:—Only the manual shut-off valves routing the fluid through the coil pair in the desired direction were initially open. The fluid temperature and rate of flow were brought to their predetermined values by making necessary adjustments while circulating the fluid through the appropriate common circuit and corresponding coil bypass line. These conditions first being satisfied, the coil pair was then put into operation by simultaneously closing the bypass line solenoid valve and opening the inlet solenoid valve to the coil.

Prior to placing a coil pair into operation by the foregoing procedure, soil temperature measurements were made to obtain the before-test distributions in the soil surrounding the coil. The appropriate recording potentiometers were switched on immediately before beginning fluid circulation to record the initial temperatures of all thermocouple junctions located about the coil pair. Each potentiometer then remained in operation for the duration of the test run and the soil temperature recovery period after the end of the run.

Soil moisture content determinations were made also just prior to the beginning of fluid circulation in each test. Borings in the ditch backfill vertically above

each leg of the coil pair to be tested were made using a hand auger of either 1-in. or 2-in. diameter. Soil samples were taken at each ½-ft increment to a depth of 5 ft (coil depth). Longitudinal spacing of the borings above the operating coils was 18 in. These borings were continued daily through the duration of the test and soil temperature recovery period.

Additional borings were made before test, 2 ft to either side of the coil leg in the natural soil. The 2-in. auger was used for the side borings which were to a

TABLE 1—SUMMARY OF SIMULATED CONDENSER TYPE RUNS

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9) ULTIMATE RATE OF HEAT TRANSFER BTU/HR	
RUN NO.	DAYS FLUID CIRCULATED	DATE FLUID CIRCULATION STOPPED	AVG FLUID INLET TEMP °F	AVG FLUID TEMP °F	AVG REYNOLD'S NO.	AVG TEMP DIFF, °F	AVG SOIL THERMAL COEFFICIENT BTU/HR (SQ FT) (°F/FT)	PER FT	°F PER FT
FE1H-I									
Case I	4.00	7 Jul 52	117.5	106.3	5443	26.4	0.79	64.0	2.42
Case II	2.00	9 Jul 52	118.4	108.0	5507	27.0	0.79	54.0	2.00
Case III	1.00	10 Jul 52	118.4	109.3	5012	28.3	0.78	42.0	1.48
FE1H	7.00	4 Sep 52	123.3	119.3	6861	32.0	0.77	15.3	0.48
FE2H	7.00	11 Sep 52	122.7	121.6	31397	35.0	0.76	15.0	0.43
FE3H	15.25	10 Oct 52	120.9	119.3	16774	35.9	0.75	18.0	0.50
1BH	17.00	11 May 50	120.0	114.8	2050	49.9	0.65	50.0	1.00
BA1H	7.00	18 May 51	130.1	115.7	2210	39.8	0.73	47.1	1.18
BA2H	7.00	30 May 51	130.1	120.2	3790	41.6	0.72	47.6	1.14
BA3H	7.00	6 Jun 51	130.1	123.8	6400	43.4	0.71	47.8	1.10
BA4H	15.00	8 Aug 51	99.3	95.9	2310	8.2	0.89	10.0	1.22
BA5H	7.00	15 Aug 51	99.5	98.4	3480	9.5	0.88	2.0	0.21
BA6H	7.00	22 Aug 51	100.4	100.0	6260	11.7	0.88	2.0	0.17
BA1H-I									
Case I	14.08	10 Jun 52	120.2	110.3	5532	36.6	0.73	95.0	2.60
Case II	5.92	16 Jun 52	120.9	114.3	5473	39.5	0.72	68.0	1.72
Case III	5.00	21 Jun 52	120.7	116.0	5414	39.8	0.72	47.0	1.18
D-1	30.00	16 Dec 49	120.0	111.2	1009	40.5	0.71	46.1	1.14
1DH	30.00	4 Mar 50	120.0	112.1	1050	50.8	0.63	57.0	1.12
1CH	20.92	27 Jun 50	120.0	115.3	1150	42.9	0.69	46.0	1.07
DC1H	5.77	12 Jun 51	130.1	117.1	1590	32.8	0.75	52.0	1.59
DC2H	7.00	22 Jun 51	129.7	122.7	2900	39.3	0.73	46.0	1.17
DC3H	7.00	29 Jun 51	129.9	126.5	5950	40.1	0.74	47.0	1.17
CD1H	9.71	9 Jul 51	130.1	121.5	1700	34.2	0.76	36.0	1.05
CD2H	7.00	16 Jul 51	130.3	124.3	3180	40.7	0.73	40.2	0.99
CD3H	15.00	31 Jul 51	129.6	126.9	6470	40.9	0.73	41.0	1.00
DC4H	16.00	7 Sep 51	100.4	96.8	1250	4.0	0.90	15.0	3.75
DC5H	17.00	24 Sep 51	109.4	107.6	2468	12.0	0.82	22.0	1.83
DC6H	21.55	10 Oct 51	110.8	109.0	5256	13.8	0.81	15.0	1.09
DC1H-I									
Case I	19.00	5 Aug 52	115.7	110.8	5979	28.8	0.78	90.0	3.13
Case II	7.08	12 Aug 52	118.4	115.3	5647	31.6	0.77	61.0	1.93
Case III	6.00	18 Aug 52	119.8	117.9	6062	32.8	0.77	35.0	1.07

depth of 6 to 8 ft. Soil samples were taken at each ½-ft increment of depth. Only one side boring per week per coil leg of the operating pair was made during the test and the soil temperature recovery period after test.

TESTS AND DATA

Of the 57 tests completed on the buried coil installation, 25 were simulated coil condenser type runs while 32 were simulated coil evaporator runs. The tests varied from 5 to 90 days duration. Both intermittent and continuous flow were

TABLE 2—SUMMARY OF SIMULATED EVAPORATOR TYPE RUNS

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)		(10)
RUN NO.	DAYS FLUID CIRCULATED	DATE FLUID CIRCULATION STOPPED	AVG FLUID INLET TEMP °F	AVG FLUID TEMP °F	AVG REYNOLD'S NO.	AVG TEMP DIFF, °F	AVG SOIL THERMAL COEFFICIENT BTU/HR (SQ FT) (°F/FT)	ULTIMATE RATE OF HEAT TRANSFER BTU/HR		MAX RADIUS OF SOIL FREEZING
								PER FT	°F PER FT	
FE1C	5.00	15 Jan 52	35.6	40.1	1698	25.7	1.00	19.0	0.74	—
FE2C	7.79	23 Jan 52	35.2	36.7	5925	29.0	1.00	21.0	0.72	—
FE3C	8.00	8 Feb 52	30.0	33.8	2206	34.0	1.00	25.0	0.74	—
FE1C-I										
Case I	4.00	12 May 52	44.6	52.3	1785	17.4	1.00	35.0	2.01	—
Case II	2.00	14 May 52	39.2	45.9	1898	24.9	1.00	32.0	1.29	—
Case III	1.00	15 May 52	41.9	47.3	1873	24.3	1.00	27.0	1.11	—
FE4C	5.00	23 Feb 53	23.2	29.3	1213	33.7	1.00	31.5	0.93	—
FE5C	5.00	28 Feb 53	20.8	24.4	2732	38.0	1.00	37.0	0.97	—
FE6C	5.00	5 Mar 53	24.8	26.6	4064	35.2	1.00	35.0	0.99	—
FE2C-I										
Case I	4.00	14 Apr 53	25.7	27.3	4391	40.3	1.00	45.0	1.12	Unknown
Case II	2.00	16 Apr 53	25.4	26.5	4446	41.7	1.00	35.0	0.84	Unknown
Case III	1.00	17 Apr 53	26.6	27.4	4590	41.1	1.00	31.0	0.75	Unknown
BA1C	9.00	2 Feb 51	33.8	40.1	734	25.0	1.00	22.0	0.88	—
BA2C	7.00	13 Feb 51	33.8	38.3	1091	23.9	1.00	20.0	0.84	—
BA3C	7.00	28 Feb 51	33.8	36.7	1640	24.7	1.00	20.0	0.81	—
BA4C	6.00	7 Mar 51	33.8	38.3	4695	25.1	1.00	20.0	0.80	—
BA1C-I										
Case I	7.00	28 Feb 52	33.1	39.4	1475	25.5	1.00	58.0	2.27	—
Case II	7.00	6 Mar 52	33.1	37.6	1442	27.3	1.00	42.0	1.54	—
Case III	7.00	13 Mar 52	33.4	36.6	1442	28.3	1.00	30.0	1.06	—
BA5C	7.00	13 Dec 52	19.9	24.8	1191	44.1	1.00	41.0	0.93	0.00"
BA6C	7.00	20 Dec 52	23.0	24.8	2876	42.3	1.00	39.0	0.92	2.50"
BA7C	10.00	12 Jan 53	24.4	25.2	5216	38.9	1.00	37.0	0.95	3.00"
BA2C-I										
Case I	8.00	30 Apr 53	22.8	24.5	4124	43.9	1.00	104.0	2.37	1.00"
Case II	7.00	7 May 53	19.4	20.8	4124	49.5	1.00	87.0	1.76	2.00"
Case III	7.00	14 May 53	21.6	22.5	4136	48.4	1.00	55.0	1.14	2.75"
1DC	8.00	21 Dec 50	33.8	34.2	245	38.3	1.00	30.0	0.78	—
DC1C	7.00	22 Mar 51	33.8	36.5	1030	28.5	1.00	26.5	0.93	—
DC2C	7.00	3 Apr 51	39.2	41.0	2560	30.2	1.00	34.8	1.15	—
DC3C	8.92	21 Apr 51	40.1	41.0	4700	29.0	1.00	36.5	1.26	—
CD1C	8.96	30 Apr 51	45.5	46.4	1240	22.8	1.00	16.2	0.71	—
CD2C	4.90	5 May 51	45.5	46.4	3090	27.8	1.00	19.2	0.69	—
CD3C	34.00	26 Nov 51	41.0	42.8	5256	20.3	1.00	30.0	1.48	—
DC1C-I										
Case I	17.00	13 May 52	35.2	40.3	1306	25.7	1.00	87.0	3.39	—
Case II	8.00	21 May 52	33.6	36.8	1323	30.6	1.00	56.0	2.18	—
Case III	6.00	27 May 52	34.2	36.7	1315	31.3	1.00	38.0	1.21	—
DC4C	10.00	20 Mar 53	26.1	27.9	1491	34.9	1.00	33.0	0.95	0.00"
DC5C	10.00	30 Mar 53	21.9	23.4	1949	41.0	1.00	31.0	0.76	2.75"
DC6C	10.00	9 Apr 53	21.9	22.5	3741	43.6	1.00	28.0	0.64	4.50"
DC2C-I										
Case I	16.00	7 Jun 53	29.1	31.9	4955	40.2	1.00	91.0	2.26	—
Case II	8.00	15 Jun 53	32.8	34.2	4890	40.9	1.00	69.0	1.69	—
Case III	6.00	21 Jun 53	38.5	39.4	4769	37.5	1.00	57.0	1.52	—
BA8C	90.88	28 Jan 54	21.2	23.0	2345	47.7	1.00	37.0	0.78	4.50"
BA8C-I										
Case I	12.00	9 Feb 54	21.7	23.7	2754	40.1	1.00	83.0	2.07	0.50"
Case II	11.00	20 Feb 54	22.1	23.2	2773	41.2	1.00	45.0	1.09	1.75"
Case III	10.00	2 Mar 54	21.9	25.5	2754	39.7	1.00	31.0	0.78	2.50"

investigated over a range of Reynold's numbers from 1000 to 6000 except for 2 runs which were below 1000 and 2 whose values were well above 6000. Fluid inlet temperatures investigated covered the range from 90 to 130 deg for simulated condenser type runs and from 20 to 45 deg for simulated evaporator runs. A mixture of ethylene glycol and water was circulated through the coils in all tests.

In Table 1 are listed all simulated coil condenser type runs with a summary of their operating conditions and results. Similarly, the coil evaporator type runs are tabulated in Table 2.

The run designation is given in column 1 of Tables 1 and 2. The initial buried coil test, a condenser type run, was designated run D-1 where D specified the coil tested and the numeral 1 indicated an initial test. Modification of the run designation was required for complete identification when the tests were changed from one coil of 75-ft length to 2 coil legs in series of 160-ft total length. Where 2 letters appear first in a run designation, those letters denote the coil pair tested, the first letter indicating the upstream leg. The numeral indicates the number of runs of that type operation involving a particular coil pair that have been made. A letter H following the numeral indicates a *hot* or condenser type run while a C indicates a *cold* or evaporator type run. Intermittent fluid flow where employed is denoted by the final letter I and a case number indicating the cycle of intermittent operation as defined earlier.

Column 5 gives the mean operating temperature of the coil during test. This temperature is the average of all arithmetic averages of the coil inlet and outlet temperatures at each computation of an instantaneous rate of heat transfer, q.

Column 6 is the computed mean Reynold's number of the circulating fluid in the coil during test. Each of the Reynold's numbers given are computed using the inside coil diameter, the average rate of fluid flow during the run, and the average fluid viscosity. The latter quantity in turn depended upon the average coil operating temperature given in column 5 and the average ethylene glycol concentration of the fluid.

Column 7 gives the difference between the mean coil operating temperature and the mean thermally undisturbed soil temperature at coil depth for the period of the test run.

In column 8 is given the mean values of a thermal coefficient of the soil immediately surrounding the operating coil. An experimental method, called the heat meter method,[1, 3, 4] was used to measure the thermal coefficients. Tests on 14 soils covering a wide range of soil types were completed in which thermal coefficients and thermal moisture migration data were obtained and correlated. The resulting correlation, was used to compute the soil thermal coefficients given in Tables 1 and 2.

In addition, column 10 in Table 2 gives the maximum radius of soil freezing in inches which was observed at the midpoint of the up-stream coil in runs where the mean fluid temperature in the coil was below 32 F.

DATA ANALYSIS

Dimensional analysis was utilized to obtain a correlation of data from the tests. Five preliminary attempts were made before an acceptable correlation evolved from the dimensional analysis study. Variables which were isolated and evaluated during a test run and which were incorporated into the final correlation were:

$$\theta, Q, \Delta T, k, \rho C_p, L, D.$$

As applied to the work of this project the definitions of these quantities are as follows:

θ = hours of operation elapsed after the beginning of fluid circulation through a coil pair.

Q = the total instantaneous rate of heat transfer in Btu per hr between the circulating fluid and soil at time θ.

ΔT = $|t - t_r|$, the instantaneous absolute difference in Fahrenheit degree between the average fluid temperature, t, in the coil at time θ and the ambient soil temperature, t_r, at coil depth (5 ft) averaged over the entire operating period of the run.

k = a thermal coefficient in Btu per (hr) (sq ft) (deg F per ft) of the soil at the buried coil surface averaged over the entire operating period of the run.

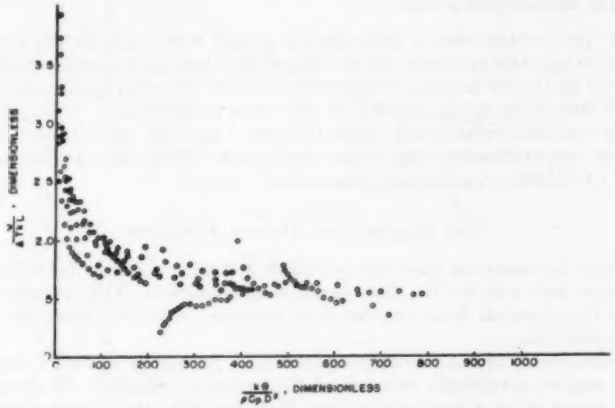

FIG. 3—DIMENSIONAL ANALYSIS CORRELATION OF DATA OF BURIED COIL CONDENSER RUNS, CONTINUOUS OPERATION

ρC_p = the volumetric specific heat in Btu per (cu ft) (deg F) of the soil at the coil averaged over the entire operating period of the run. ρ is the wet density of the soil around the coil while C_p is the wet soil heat capacity.

L = the effective length in feet of the coil or coil pair being tested.

D = the coil inside diameter, feet.

The final dimensionless grouping of the above variables was

$$(Q/\Delta T k L) \text{ and } (k\theta/\rho C_p D^2).$$

Similar groups were obtained by Hadley,[2] but his definitions of certain of the variables are different from the definitions used here. Differences occur in the definition of the rate of heat transfer, Q. the temperature difference, ΔT, and the time, θ.

Fig. 3 is a plot of group $(Q/\Delta T k L)$ versus group $(k\theta/\rho C_p D^2)$ for all continuous operation simulated condenser type runs. The ordinate of this curve decreases monotonically. The large initial values of the ordinate corresponded to high rates of heat transfer immediately after beginning fluid circulation through the coil

pair. An approach to stabilized soil temperature and heat flow conditions was observed as the abscissa increased. The steady-state conditions approached are represented in Fig. 3 by the level portion of the data.

If only the horizontal part of the data in the plot is considered, then the ordinate is independent of the group $(k\theta/\rho C_p D^2)$, or

$$Q/\Delta T k L = I \quad \ldots \quad (1)$$

Rearranging the expression, the effective buried coil length is found to be

$$L = Q/\Delta T k I \quad \ldots \quad (2)$$

where

I, called the ***intermittency factor,*** is a function of the fraction of unit time that fluid is circulating through the coils and is equal to the level value of ordinate approached in the plots of dimensionless groups.

Similar plots of the same 2 dimensionless groups were made for the continuous simulated evaporator type runs and for each of the 3 cases of intermittent operation. Equation 1 was found to apply in every instance with the value of the intermittency factor, I, increasing as the fraction of unit time *on* decreased. Figs. 4a and 4b show the empirical relationship found between I and the intermittency ratio for condenser and evaporator type runs, respectively. Note that an intermittency ratio of 1.0 represents continuous operation.

THE BURIED COIL DESIGN EQUATION

Equation 2 formed the basis around which a design procedure for a buried coil heat source and sink for the heat pump was developed. The expression is the result of experimental data compiled from tests on 3 relatively short lengths (160 ft) of buried coil.

The variables Q and ΔT were computed from quantities measured during test while I was an empirically determined dimensionless number. Evaluation of k was by means of an equation which resulted from laboratory tests of artificially prepared soil samples. The equation for k given here without derivation[1, 3, 4] is

$$k = a\left(\frac{t}{t_r}\right)^{\frac{0.29}{d+1}} (S_r)^{0.29} \quad \ldots \quad (3)$$

where

k = soil thermal coefficient corresponding to the stabilized soil condition established during coil operation.
S_r = initial or reference degree of saturation of the soil.
t = average fluid temperature in the coil, Fahrenheit degrees.
t_r = average ambient soil temperature at coil depth, Fahrenheit degrees.

and a and d = empirical constants.

If the expression for k given by Equation 3 is substituted into Equation 2 together with $\Delta T = (t - t_r)$, the final form of the design equation is

$$L = \frac{Q}{\left[a\left(\frac{t}{t_r}\right)^{\frac{0.29}{d+1}} (S_r)^{0.29}\right](t - t_r)I} \quad \ldots \quad (4)$$

Thus, a buried coil for the heat pump may be designed from a knowledge of the total coil load, Q, the mean coil operating temperature, t, a reference soil temperature, t_r, the initial degree of saturation of the soil at coil depth, S_r, and 3 empirical constants, a, d, and I.

The design consists of an evaluation of these quantities followed by a solution

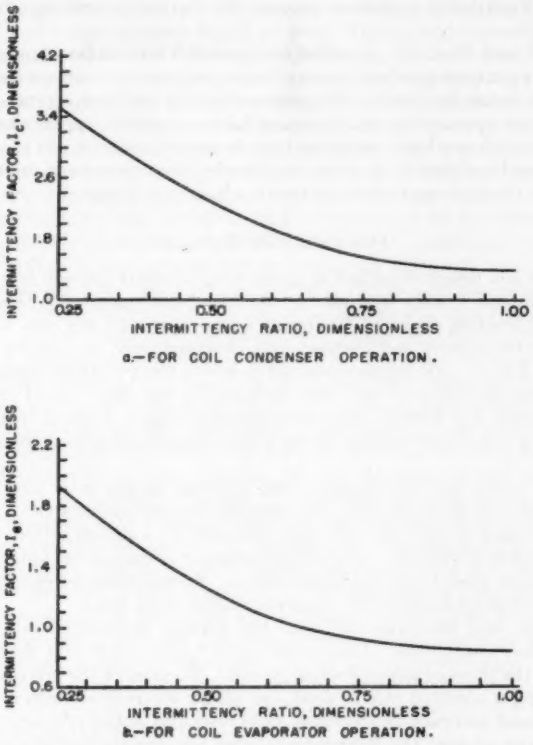

FIG. 4—EMPIRICAL RELATIONSHIPS OF THE INTERMITTENCY FACTOR WITH THE COIL INTERMITTENCY RATIO

of the equation for the coil length L. This length must then be modified to suit the particular coil configuration employed[1].

BURIED COIL CONFIGURATIONS

In practice, the ground surface area under which a coil is to be buried may be too small to permit installation of a coil arrangement where each leg is thermally iso-

lated from the others. A solution to the problem would be to space adjacent coil legs closer than the minimum separation required for thermal isolation. In this case the influence of each leg upon adjacent legs will reduce the rate of heat transfer between soil and fluid per linear foot of buried coil, thus necessitating a greater coil length than that given by Equation 4 to meet a specific load. No experimental investigation of thermal interference was attempted but a theoretical analysis of the problem described in the Appendix served as the basis for including in Equation 4 a ratio to account for thermal interference between coil legs.

Values of R and R' which are called configuration factors (see Appendix for the derivation of equations for these factors) were computed for various configurations and are given in tabular form in the proposed buried coil design procedure[1]. The reciprocal of the appropriate configuration factor, depending upon whether one or two coils per ditch are used, multiplied by design Equation 4 will result in a coil length increased sufficiently to make up for the heat transfer reduction due to the effects of thermal interference between adjacent coil legs.

DISCUSSION OF RESULTS

The buried coil design equation as given is valid where the soil is unfrozen and also where the soil freezes to some extent during coil operation. The maximum radius of soil freezing observed during an evaporator type run was 4.5 in.

Effects attributable to soil freezing were observed only for a short time at the beginning of a test. All evaporator runs in which the operating temperature was below the freezing point of soil were included in the dimensional analysis plot. Data from these soil freezing runs corresponded closely to data from nonfrozen runs following the initial period during which the radius of soil freezing was increasing.

It may be noted that coil diameter and material do not enter into Equation 4. Data from tests of all 3 coil sizes were included in the final analysis. No distinction between data of different coil sizes was evident, indicating that the effect of coil diameter was negligible. The design equation is considered applicable, therefore, to coil sizes from ½ to 2 in. in diameter. No coil diameter outside this range and no coil of material other than copper was used in the tests.

The rate of fluid flow was such that the average Reynold's number exceeded 1000 in all but 2 test runs. Data from these 2 runs, 1DC and BA1C, were excluded from the plots of dimensionless groups. Because of the increased difficulty in maintaining a constant rate of fluid flow at low Reynold's numbers and in view of the decreased accuracy in recording fluid inlet and outlet temperatures at relatively low rates of flow, the Reynold's number was maintained above 1000 for all other tests. This value is recommended as the minimum to be used in coil design where the data from this work are used.

It must be emphasized that only one fluid was circulated through the buried coils in all tests. A liquid solution of ethylene glycol in water was used exclusively, and the solution experienced no change of phase while traversing the coil. For the results of this work to be strictly applicable to a design problem, the fluid circulated through the coil must be a liquid.

Justification for applying the data to the flow of a boiling or condensing liquid refrigerant as would be the case in a direct expansion system was found in a study of the coil internal and external heat transfer film coefficients.[5,6,7,8] In the coil tests studied where liquid was circulated, the ratio of the inside to outside film co-

efficient was greater than 10. This difference in magnitude rendered negligible the influence of the inside film coefficient upon the overall heat transfer coefficient. Further, for boiling or condensing dichlorodifluoromethane the average inside film coefficient was computed by the method of Baker, Touloukian and Hawkins[6] and was found to be larger than those determined for the water-ethylene glycol solution assuming the same rate of heat flow per unit inside surface area for both types of flow.

These findings show that the results of this work may be applied to the design of a buried coil where either a liquid or where boiling or condensing dichlorodifluoromethane is circulated. The foregoing analysis also confirmed the fact which was indicated by the dimensional analysis study that no upper limit of Reynold's number should be placed on the data. The inside film coefficient for the types of flow discussed normally increases with Reynold's number. An increase in Reynold's number would further decrease the already negligibly small influence of the inside film coefficient upon the rate of heat transfer between the fluid and surrounding soil.

The absolute difference, ΔT, between average coil temperature and ambient soil temperature should be maintained at 25 F or greater in an installation. During runs BA4H, BA5H, BA6H, DC4H, DC5H, and DC6H in which ΔT was well below that value, difficulty was encountered in measuring the small drop in fluid temperature across the coil. Very erratic heat flow values were computed for those 6 runs, and their data were omitted in the final dimensional analysis plot.

Reliability of the result from the buried coil design equation depends upon the evaluation of the factors applying to a particular design problem and also on the accuracy of the measured data used in the dimensional analysis correlation.

An analysis was made of the errors involved in measuring the various quantities in design Equation 2. In the observation of data during a buried coil test, seemingly small inherent instrument errors were encountered in the measurement of coil inlet and outlet fluid temperatures. All thermocouples at the field site were calibrated before installation in a constant temperature bath, the temperature of which was measured with a mercury-in-glass thermometer of known calibration. It was found that the differences, Δt, between junction temperatures of coil inlet and coil outlet thermocouples could be measured correctly to $\pm$ 0.18 F (or $\pm$ 0.1 C) at the temperature levels employed in the coil testing program. The estimated percentage error in measuring Δt through all the field runs, based on $\pm$ 0.18 F was $\pm$ 4.0 percent. And the percentage error would appear to be no greater than $\pm$ 2 percent in each measurement of the instantaneous rate of fluid flow. For the majority of runs an even smaller error was involved.

Assuming that the error in the value of fluid heat capacity, c, may be neglected, and by examining the equation

$$Q = w\, c\, \Delta t$$

it may be seen that an error in each computation of instantaneous rate of heat transfer, Q, was $\pm$ 6 percent.

If Q is correct to $\pm$ 6 percent, ΔT is correct to $\pm$ 4 percent, and k and L are considered without error, then the estimated error in the dimensionless group $Q/\Delta TkL$ is $\pm$ 10 percent. The analysis is applicable to intermittent as well as continuous coil operation. In obtaining the basic design Equation 2, the dimensionless group $(Q/\Delta TkL)$ was equated to a constant, which was called the intermittency factor I. Thus, the error in I for any one test run is identical to that for the group. An independent check of the average deviation of I from a mean value

for all continuous operation runs was computed to be ± 9 percent. As the fraction of coil operation per unit time decreased this average deviation increased.

For design purposes, the values of I plotted in Figs. 4a and 4b were selected with the intention that the resulting computed coil length be adequate during periods of extreme loads without auxiliary equipment to provide supplementary heating or cooling. The values of I plotted were the most reasonable values near the lower limit of the range of I corresponding to each type of test and cycle of operation.

Coil design for extreme ambient temperature conditions was the primary consideration in evaluating the reference soil temperature. An investigation of published weather station data indicated the greatest need for cooling occurred consistently in the months of July and August and the largest heating requirements were in January and February. The most unfavorable soil temperature conditions were found to occur in the months of August and February.[3] Thus, to perform satisfactorily during those extreme periods the reference soil temperature, t_r, was taken as the mean soil temperature at coil depth during August and February for coil condenser and evaporator operation, respectively.

The development of Equation 2, which resulted from the dimensional analysis of the operating coil data, into the usable buried coil design Equation 4 involved data correlations from laboratory soil heat transfer tests and thermal moisture migration phenomena investigations. These correlations were expressed in Equation 3 for the evaluation of a soil thermal coefficient in terms of the initial soil condition, the ratio of mean coil operating temperature to a reference soil temperature, and 2 constants. The reliability of Equation 3 in actual practice depends largely upon the accuracy with which the properties of the soil at coil depth are known. Emphasis is placed on the fact that standard procedures should be followed by an experienced field crew where possible in making density measurements and in obtaining soil samples at the proposed buried coil site. It is recommended that the soil index properties, liquid limit and plastic limit, which are related to the constants a and d be measured by experienced soil technicians[1].

The refrigeration machine condenser and evaporator capacities are required in the design equation. Also, the heat loss during the winter season and the heat gain during the cooling season of the space to be conditioned must be computed. The procedures recommended to find the heat loss or gain are those of ASHAE. Values determined by these methods, in general, are conservative. Manufacturers ratings at specified operating conditions of a given piece of equipment are normally *safe* values to use in sizing equipment for the job. In actual practice, however, variations in operating characteristics will occur causing a corresponding change in equipment capacity.

Because the accuracy of various site and operating equipment quantities necessary to the design of a buried coil cannot be estimated, the constants in Equation 4 were selected such that the coil length given by Equation 4 will tend to be slightly oversized.

As pointed out previously, no experimental investigations have been conducted to verify the configuration factors. It is thought, however, that these also will produce conservative coil lengths.

ACKNOWLEDGMENT

This paper was prepared by B. J. Fluker while D. M. Vestal, Jr. was on leave of absence. Mr. Vestal acted as consultant concerning questions which arose during the preparation of the paper.

Acknowledgment is made of the excellent foremanship of J. W. Overall, Jr., in the performance of the test work and in the collection of data described and presented in this paper.

Acknowledgment is given also to those students of the A. & M. College of Texas who assisted in all phases of work of this project.

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3. A future paper is planned which will present the natural soil temperature data collected by the project. Another paper will be prepared dealing with the soil thermal conductivity work conducted while a third paper will describe the moisture migration studies made. Data for the latter two papers were observed in laboratory tests of artificially compacted specimens covering a wide range of soil types.

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APPENDIX

DETAILS OF EQUIPMENT AND APPARATUS

Each coil pair at the field site was buried at a nominal depth of 5 ft below the ground surface in ditches approximately 20-in wide. Cross sections of ditches A and B, C and D, and E and F are given in Figs. A-1 and A-2. All the soil removed from ditches B, D, E, and F was put back in them at (as nearly as possible) its natural density and natural moisture content. Tamping was required to attain the backfill density desired in each ditch.

Special attention was given ditches A, B, and C in attempts to provide means for maintaining optimum soil moisture conditions near the buried coil. In the backfill of ditch A, 2 steel reinforcing rods were embedded one each in the geometric center of two 10-in. square cross sections of coke breeze, as shown in Fig. A-1. The rods surrounded by coke breeze extended the entire length of coil A. These rods were intended to serve as anodes in the study of moisture movement by electroosmosis. The copper coil was to be the cathode. The purpose of this electroosmosis set-up was to provide a means whereby moisture which migrated from the coil due to a temperature difference could be forced back or held motionless by an opposing electrical potential. The balance between these 2 forces was to be studied and the thermal moisture migration analyzed with the aid of known information about electroosmosis.

In ditch B a ½-in. diameter tracer pipe was installed with the coil as shown in Fig. A-1. The tracer pipe extended the entire length of coil B and was perforated at uniform

intervals along its length. One end of the horizontal pipe was joined to a vertical length of pipe which extended above the ground surface and was covered with a removable cap. The purpose of the tracer pipe and vertical riser was to permit addition of water to the soil near the buried coil surface to maintain the most favorable soil heat transfer characteristics possible.

The electroosmosis and tracer pipe studies were discontinued after only preliminary testing because of the unexpected absence of well defined soil moisture content variations near the coil. This was thought to be due to the lack of satisfactory moisture content measuring devices with which to measure the variations and to the particular natural soil conditions at coil depth.

No data and no discussion is given covering the effects of electroosmosis and artificial water addition, since no complete investigation was made in either case and no con-

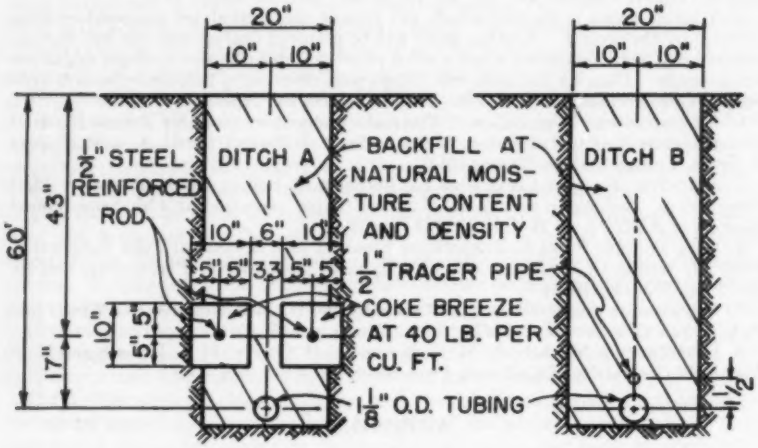

FIG. A-1—CROSS SECTION OF DITCHES AND COILS *A* AND *B*

clusions were reached concerning their influence upon the soil heat transfer characteristics.

A portion of the natural soil in ditch C was enclosed in a moisture proof barrier as shown in Fig. A-2. The barrier extended the entire length of coil C and was closed and moisture proofed also on each end. The natural soil inside the barrier was backfilled at its moisture equivalent and the barrier sealed. All other backfill in ditch C above the moisture barrier was near natural moisture content and natural density. All tests of coil C were made with this arrangement.

A system of thermocouples was placed in a concentric pattern about the coil to measure soil temperatures near the longitudinal midpoint of each of the coil legs A, B, C, and D. Thirty-three thermocouples were used in the cross section. The typical procedure employed to install each thermocouple in its predetermined position underground is shown in Fig. A-3. The location plan followed in placing each junction around a coil leg is given in Fig. A-4. A total of 80 thermocouples, 2 inside each of the 2 legs, 5 on the outer surface of each leg, and 33 in the soil around each leg were installed with the 1-in. coil pair and a like number with the 2-in. coil pair.

Another plan was developed and followed in locating 6 thermocouples in the soil at coil depth near coils E and F, 8 thermocouples on the outer surface of these coils, and 2

thermocouples inside the coil. A plan view of this coil pair is presented in Fig. A-5 showing the location of each of the 16 thermocouple junctions.

Lead sheathed thermocouple extension wire, No. 16 B & S gauge, was brought near to the ground surface from each thermocouple junction. At that point No. 30 B & S gauge, enameled, single cotton thread covered wire was soldered to the larger wire to make up the thermocouple extensions from a point very near the ground surface above the junction location to the temperature recording potentiometers located inside the instrument house. Three 16-point recording potentiometers were used to record continuously 176 different temperatures in, on, and around the various coils.

Two separate flow circuits located in the instrument house were used either to heat or to cool the fluid flowing through a buried coil during test. For simulated coil condenser

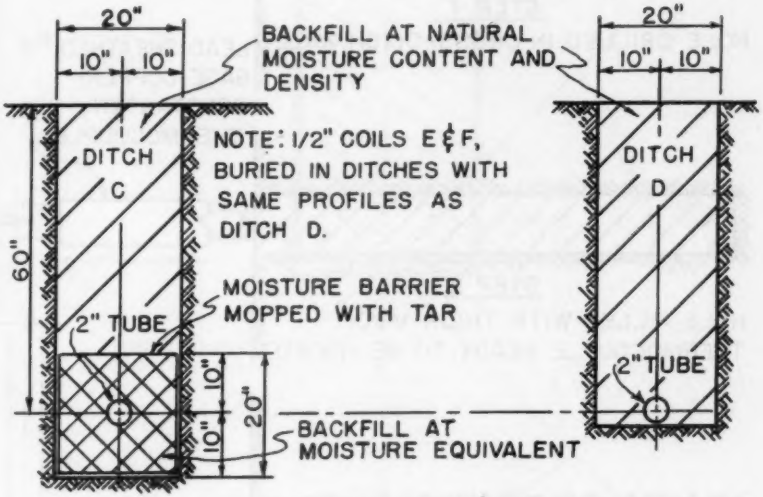

FIG. A-2—CROSS SECTION OF DITCHES AND COILS *C* AND *D*, *E* AND *F*

type runs, the fluid was routed through the common hot circuit and heated by passing it through an 8-in. diameter transite pipe approximately 5 ft in length which contained four 750-watt immersion type heaters. Control of the fluid temperature at the transite pipe outlet by an immersion type thermostat in effect controlled the fluid temperature at the inlet to the buried coil. A 5-hp water cooled condensing refrigeration unit with a tube-in-shell flooded evaporator provided fluid refrigeration for coil evaporator type runs. The fluid temperature at the coil inlet was controlled by a thermostat located in the fluid outlet side of the refrigeration system flooded evaporator. The fluid flow through a coil was maintained at a constant rate throughout a test by a float valve mechanism regulating the head on a V-notch weir. A separate fluid flow control weir was located in either the common hot circuit or the common cold circuit.

Sufficient valves were installed in the circulating fluid flow system to permit any one of the 3 coil pairs to be selected for testing. Either simulated condenser or simulated evaporator type operation could be chosen and either direction of fluid flow through the coil pair could be employed. The fluid which was circulated through the hot or cold

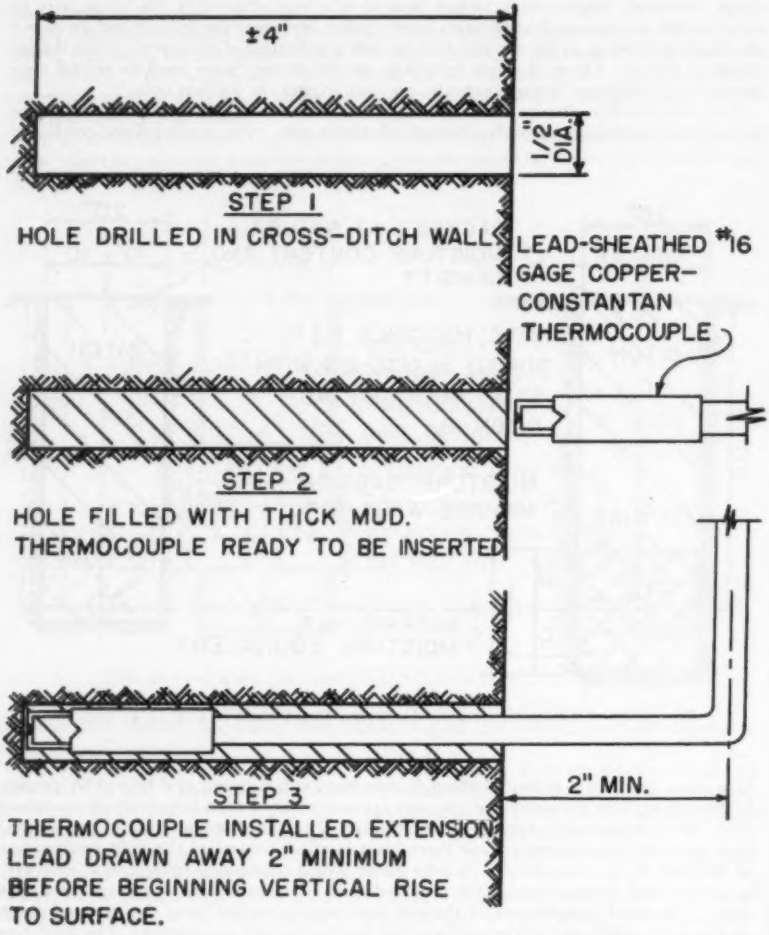
±4"
1/2" DIA.
STEP 1
HOLE DRILLED IN CROSS-DITCH WALL
LEAD-SHEATHED #16 GAGE COPPER-CONSTANTAN THERMOCOUPLE
STEP 2
HOLE FILLED WITH THICK MUD.
THERMOCOUPLE READY TO BE INSERTED
2" MIN.
STEP 3
THERMOCOUPLE INSTALLED. EXTENSION LEAD DRAWN AWAY 2" MINIMUM BEFORE BEGINNING VERTICAL RISE TO SURFACE.

FIG. A-3—TYPICAL THERMOCOUPLE INSTALLATIONS FOR THE MEASUREMENT OF SOIL TEMPERATURES

common circuit and any coil pair, consisted of a solution of ethylene glycol in water. The percentage of ethylene glycol in solution averaged 21.5 percent by weight over the entire operation of the buried coils.

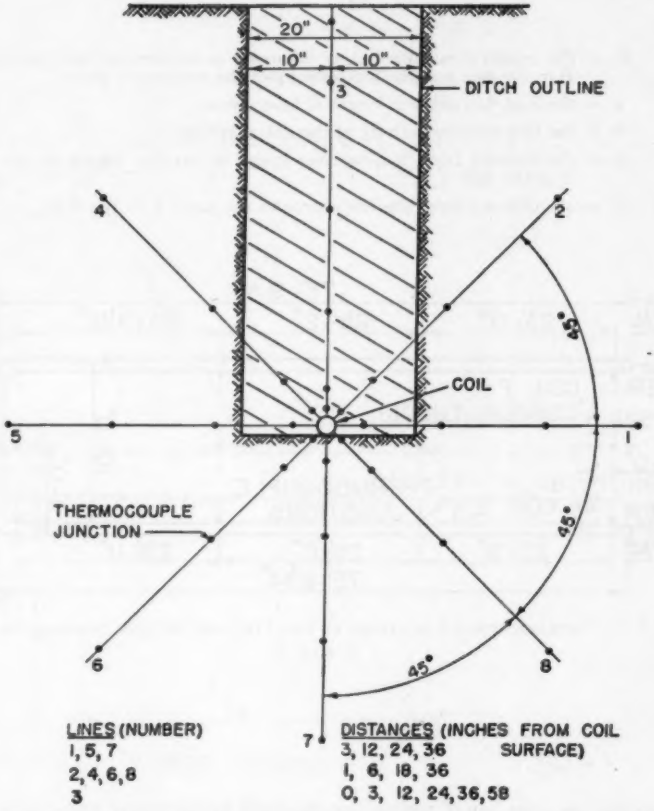

FIG. A-4—CROSS SECTION OF THERMOCOUPLES IN THE SOIL AROUND THE MID-POINT OF A BURIED COIL

THEORETICAL ANALYSIS OF THERMAL INTERFERENCE BETWEEN COIL LEGS

Application of design Equation 4 to non-isolated coils involved an adaptation of the method of super-position. Bridgers (reference A-1) arrived at an equation expressing the ratio of the strength of one of a series of buried coils affected by thermal interference to the strength of the same coil isolated at the same temperature. It was shown by Durand (reference A-2) that potential fields can be combined by adding the potentials

of the individual fields. The temperature potential function (references A-2, A-3, A-4) of a line source in an infinite slab bounded on one side by an isothermal plane has been found to be

$$\theta_1 = \frac{q}{2\pi k} \ln \frac{r_p}{r_c} = \frac{q\gamma}{2\pi k} \quad \text{. . . . . (A-1)}$$

where

θ_1 = the temperature potential, or difference in temperature between a point P in the slab and the temperature of the isothermal plane

q = the heat flux per unit length of heat source

k = the thermal conductivity of the slab material

r_p = the distance from the negative image of the line source to the point P in the slab

r_c = the distance from the line source to the point P in the slab,

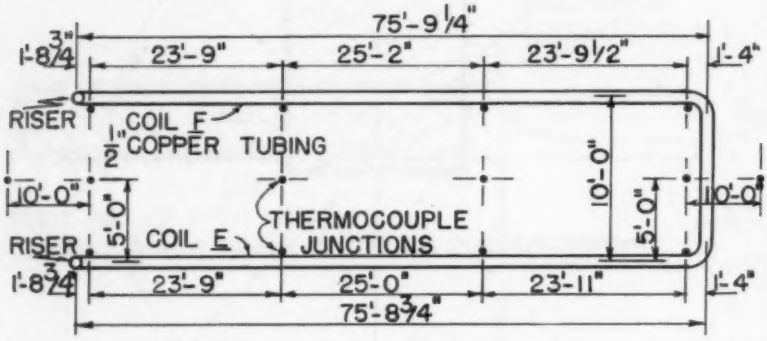

FIG. A-5—THERMOCOUPLE LOCATIONS IN THE SOIL AND ON THE SURFACE OF COILS *E* AND *F*

and

$$\gamma = \ln \frac{r_p}{r_c}.$$

Applying the procedure of adding potential fields in the case of 2 sources in a slab with the point P on the surface of one of the sources as illustrated in Fig. A-6, Bridgers developed the equation

$$N = \frac{\theta_0}{\theta_0 + \theta_1} \quad \text{. . . . . (A-2)}$$

where

N = the strength ratio, defined as the ratio of heat dissipated by one of 2 embedded sources both at equal temperature to the heat dissipated by only one thermally isolated source at that same temperature

θ_0 = the temperature potential or difference between the temperature of the embedded source and that of the isothermal plane

while

θ_1 = the temperature potential determined by Equation A-1.

Similarly, the strength ratio of any single source of a series of n heat sources of known spacing is

$$N = \frac{\theta_0}{\theta_0 + \theta_1 + \theta_2 + \cdots + \theta_n} \quad \text{(A-3)}$$

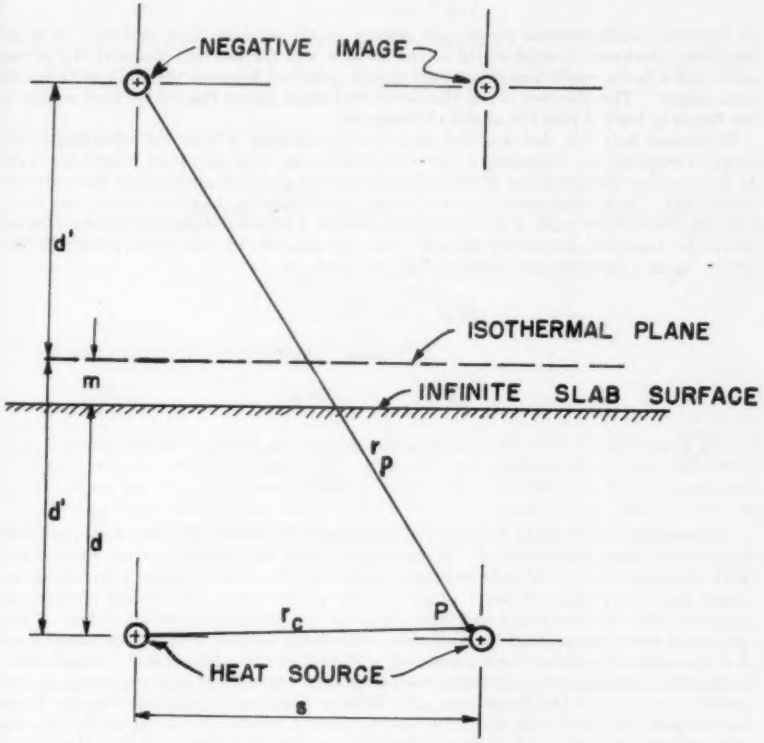

FIG. A-6—COMBINATION OF TWO HEAT SOURCES BURIED IN AN INFINITE SLAB

The idealized problem considered was one in which a number of line sources or sinks is buried in an infinite slab bounded above by an air-cooled isothermal plane. If the source diameter is small compared with its depth of bury, d, and center-line spacing, s (see Fig. A-6), then the ratio r_p/r_c required in Equation A-1 may be computed by

$$\frac{r_p}{r_c} = \left[1 + \left(\frac{2d'}{s}\right)^2\right]^{1/2} \quad \text{(A-4)}$$

The quantity d' is the distance between the source and the isothermal boundary. Schofield (reference A-3) points out that the surface of the solid medium, the ground surface in this case, will not be the isothermal boundary, since the solid surface temperature distribution is governed by the balance at each point between the heat conducted to the surface and that transferred away from the surface by convection and radiation. The isothermal, therefore, exists in the atmosphere at some distance above the surface of the solid medium. Schofield (reference A-3) developed the equation

$$m = \frac{k}{h} \quad \text{(A-5)}$$

for locating the isothermal plane with respect to the solid medium surface. m is an imaginary thickness of solid added at the surface, k is the thermal conductivity of the solid, and h is the coefficient of convection and radiation between the solid surface and atmosphere. The distance, d', of the isothermal plane above the buried heat source is the depth of bury d plus the added thickness m.

Equations A-1, A-2, A-4 and A-5 were used to develop a basis for adjusting a coil length computed by Equation 4 for either source or sink operation (reference A-2) to account for the reduction of heat transfer due to thermal interference between adjacent legs. It is anticipated that the range of coil spacing may vary from 5 to 10 ft, coil depth may vary from 3 to 6 ft, and more than 2 horizontal parallel legs will be required for workable buried coil designs. For the general case, the strength ratio of the center leg of 3 parallel legs (single coil leg per ditch) is

$$N = \frac{1}{1 + \frac{q\gamma}{2\pi k\theta_0} + \frac{q\gamma}{2\pi k\theta_0}}$$

or

$$N = \frac{1}{1 + \frac{q\gamma}{\pi k\theta_0}} \quad \text{(A-6)}$$

Theoretically, a strength ratio of 1.0, corresponding to thermal isolation, cannot be obtained by using Equation A-6. Examination of the soil temperature gradients which were observed during coil tests indicated, however, that little thermal interference between legs was evident in the U-shaped configuration used. Horizontal temperature gradients were observed to a radius of 3 ft from the midpoint of each coil leg. These gradients were extrapolated on both sides of each leg and the temperatures of the soil 5 ft from the coil surface were compared with that of the ambient soil. Small differences were found to exist which may well have been normal due to variations in ground surface cover above the 2 coil legs. No definite trends or appreciable changes in the soil temperature due to coil operation were found at a location 5 ft horizontally from the coil surface. For practical purposes, then, it was assumed that a spacing of 10 ft between adjacent coil legs was sufficient to maintain a coil strength closely approaching that of a thermally isolated coil. Strength ratios for all other configurations are based upon that for the coils tested.

Coil lengths of configurations with $N > N_r$ where N_r is a reference strength ratio computed for the spacing and depth dimensions of the test coils, will require no correction due to thermal interference. Coil configurations with values of $N < N_r$ will require an adjustment of the coil length computed by Equation 4.

If the connecting crossline effects at one end of the U-shaped coil arrangement are neglected, the strength ratio, N_r, for the test configuration may be computed by Equation A-6. For a ground surface, m is approximately 0.5 ft, and d' is then 5.5 ft while

$r_0 = 10$ ft. The ratio r_p for these dimensions by Equation A-4 is 1.49. Evaluation of Equation A-1 for θ_1 and substitution into Equation A-2 gives

$$N_r = \frac{\theta_0}{\theta_0 + \frac{q}{2\pi k}} \quad (0.4)$$

or

$$N_r = \frac{1}{1 + \frac{q}{\theta_0 k}\left(\frac{0.2}{\pi}\right)}.$$

Examining coil design Equation 1, it is found that

$$\frac{Q}{kL} = \frac{q}{k} = I(\Delta T) = I(t - t_r).$$

Then, noting that $\theta_0 = (t - t_a)$,

$$\frac{q}{\theta_0 k} = I\left[\frac{t - t_r}{t - t_a}\right] = I\beta$$

where t_a is air temperature and $\beta = [(t - t_r)/(t - t_a)]$. By substitution, Equation A-6 becomes

$$N_r = \frac{1}{1 + \left(\frac{0.2}{\pi}\right) I\beta} \quad \ldots\ldots\ldots \quad (A\text{-}7)$$

Dividing Equation A-6 by Equation A-7, gives

$$R = \frac{N}{N_r} = \frac{\pi + 0.2\, I\beta}{\pi + I\beta\gamma} \quad \ldots\ldots\ldots \quad (A\text{-}8)$$

It has been found desirable in practice and in certain other investigations to place 2 coil legs in the bottom of the same ditch. The foregoing analysis shows that the rate of heat transfer per foot of coil length will be appreciably decreased by this arrangement, but the resulting combined strength of the pair will be greater than that of only one coil leg. The strength of each leg in this arrangement is $q' = Rq$ where

$$R = \frac{\pi + 0.2\, I\beta}{\pi + \frac{I\beta\gamma'}{2}}, \quad \ldots\ldots\ldots \quad (A\text{-}9)$$

and q is the rate of heat flow per foot of isolated coil. γ' corresponds to the spacing between coil legs in the same ditch. Two legs per ditch may then be replaced by only one of equivalent strength $2q' = 2Rq$. The reference coil strength for this case is considered to be $(2Rq)$. The configuration factor for 2 coils per ditch is:

$$R' = \frac{\pi + (2R)\, 0.2\, I\beta}{\pi + (2R)\,\frac{I\beta\gamma}{2} + (2R)\,\frac{I\beta\gamma}{2}}$$

or

$$R' = \frac{\pi + 0.4\, IR}{\pi + 2\, IR\beta\gamma} \quad \ldots\ldots\ldots \quad (A\text{-}10)$$

Thermal interference upon the center leg by any coil legs other than those immediately adjacent to the center one is assumed to be negligible. This conclusion was based on the results of the observed horizontal temperature gradients which were discussed

earlier in this section. These gradients indicated only small temperature changes in the soil as far distant as 5 ft horizontally from a coil surface. Inasmuch as the minimum spacing anticipated between adjacent coil legs is 5 ft, superimposed thermal interference effects from sources twice removed (10 ft or greater between the coil legs in question) will be identical to or less than those encountered during the actual coil tests. Since the test coil strength ratio, N_t, was established as a standard, ratios greater than N_t

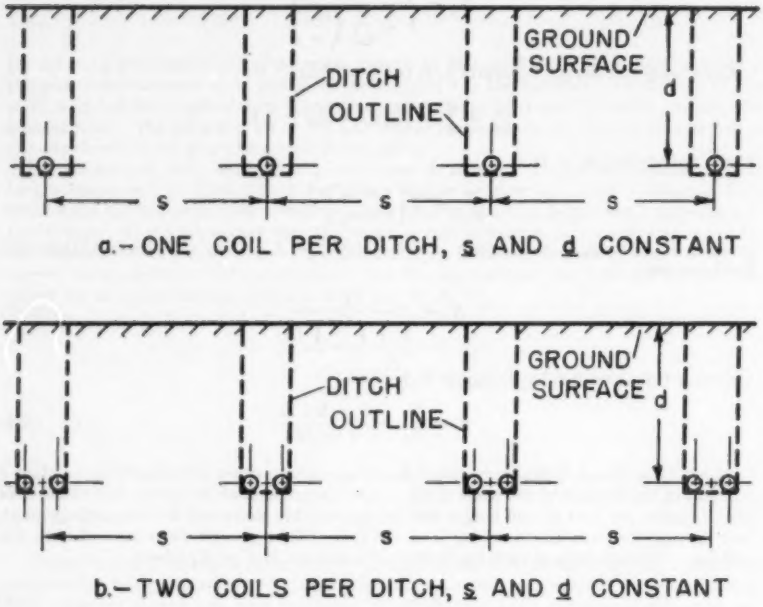

FIG. A-7—COIL CONFIGURATIONS USING ONE OR TWO COILS PER DITCH

require no correction of length due to thermal interference, the effects from legs spaced greater than 10 ft away may be neglected.

R and R' are called configuration factors. R is applicable to the case where only one coil leg per ditch is used, the spacing, s, between adjacent ditches is equidistant, and the depth of bury, d, is constant for all legs such as is shown in Fig. A-7a. R' is applicable where the ditch spacing, s, the coil leg spacing, s', and the coil depth, d, are each constant as illustrated in Fig. A-7b.

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DISCUSSION

WILLIAM L. MCGRATH, Syracuse, N. Y. (WRITTEN): This paper appears to add important experimental results to the background of information on earth source heat pump installations.

Of considerable importance in the evaluation or use of such data is knowledge of soil conditions which seems to be lacking in this paper. Particularly the moisture content of the soil as it affects both conductivity and heat storage capacity is essential to interpretation and use.

I would recommend that the authors consider this point and supplement the record with this additional information.

MERL BAKER, Lexington, Ky. (WRITTEN): Dr. Vestal and Mr. Fluker are to be congratulated on the culmination of an extensive research program. This problem is one in which the influencing factors are difficult to control and it is not easy to clearly discern the relationship among these even after extensive experiments.

It is known from both theory and experiment that the heat strength of a tube buried in the earth is a function of the undisturbed soil temperature, the thermal conductivity of the soil, the thermal diffusivity of the soil and the temperature difference between the tube surface and the undisturbed soil. The physical properties of the soil vary widely

with moisture content and the undisturbed earth temperature is a seasonal function. These variations are undoubtedly responsible for the necessity of introducing the empirical factors into the equations of this paper.

The use of dimensional analysis in correlating the experimented results is highly recommended and the authors have established a satisfactory correlation of their data. It is my considered opinion that similar correlations involving the fundamental factors, such as the thermal diffusivity and its dependence on moisture content must be introduced to render any design equation universally acceptable. The contributions given in this paper are indeed important and I would be interested in learning the future research with buried tube heat exchangers that the authors anticipate.

J. V. BORRV, Minneapolis, Minn. (WRITTEN): There has been considerable work done on the various facets of the problem of heat transfer to and from pipes surrounded by various media. However, a relatively small amount of these efforts has been directed toward an equation or equations which could be used by the mechanical designer. I am always very much in favor of projects and presentations that tend to help bridge the gap between research and practical application.

Upon reading the paper I find 6 points that might bear further clarification or investigation:

1. In Fig. 3 there appear to be 3 or 4 distinct *chains* of data points that seem to form comparatively smooth curves among the general scattering of points from the various runs. Do each of these *chains* perhaps represent the data from a particular run, and, if so, is there any explanation of why the one in the lower center of the graph shows a general increasing characteristic, while the over-all nature of the curve is to decrease with time?

2. The authors have presented quite a bit of reasoning to justify the use of an ethylene glycol solution circulating as a single state mixture instead of a refrigerant going through its normal 2-phase cycle. I believe that the reasoning was very sound as far as it went. I should like to mention, however, that when using a refrigerant the suction pressure and temperature will drop as the temperature of the heat source, in this case the ground, drops. We also know that the head pressure and temperature will rise as the temperature of the sink, also the ground in this case, rises. This results in varying inlet and outlet coil temperatures and tends to maintain a more constant temperature differential between the coil and the ground. Therefore, with a refrigerant the steady state or constant abscissa part of Curve 3 might occur at a higher capacity both for the heating and the cooling runs.

3. There is another phenomenon of interest concerning soil conductivities. Most soils at any given normal moisture content show a marked increase in conductivity when going from the unfrozen state to the frozen state. I believe that this is of considerable importance especially in a more severe climate such as ours, where heating is of major importance and soil freezing around an evaporator coil would probably be quite pronounced. I could not tell from the presentation whether or not this condition was covered by Equation 3 since a derivation was not given.

4. I think that a suggestion for the use of Equation 4 would be to take s_r as the minimum average degree of saturation for the locality in question because the degree of saturation is probably the greatest single factor affecting soil conductivities.

5. If I correctly understand the significance and use of the intermittency ratio and intermittency factor they bring to light an operational characteristic of ground coils that is rather surprising to me. That is, for a condensing run if the operating time is doubled from 50 percent to 100 percent, the total capacity for any given interval of time, say 10 hours, is increased only 24 percent. The same increase in percent running time for an evaporating run increases the total output only 36 percent. I was aware that the ratio between percent running time and total capacity would be far from 1:1, but I did not realize that it would be as high as 3:1 or 4:1.

6. I believe that the value of this paper could have been greatly enhanced by the inclusion of more original data from the various runs, typical values for the empirical constants, sample problems using the various equations, and a clearer explanation of the relationship and derivation of the various equations.

F. R. O'BRIEN*, Birmingham, Alabama (WRITTEN): Several years ago I had the opportunity to inspect the test installation discussed in this paper. I was certainly impressed with the detailed planning of the test program and the care taken in amassing data, some of which is presented here.

In December 1949, Dr. W. A. Hadley presented in the *Edison Electric Institute Bulletin* a rational correlation of the operating characteristics of heat-pump ground coils. This correlation was based on the same basic set of dimensionless groupings of variables as suggested now by the authors although as indicated in this paper there are some differences in the definitions of the variables.

I am struck with the apparent good correlation between Dr. Hadley's presentation and the curves of Figs. 3 and 4. If it is considered that Dr. Hadley's data is based on heat pumps operating with an intermittency ratio close to 1, and it is assumed that the differences in variable definitions are related only to time, then Vestal and Fluker have corroborated the earlier study very well.

The authors must be credited with the extension of the relationship $Q/\Delta TkL$ into a design equation with the incorporation of the intermittency factor I. I do question somewhat the elaborateness of the relationship given for k.

Study of this and other papers dealing with the ground coil seems to indicate that the rule-of-thumb 300 linear feet of ground coil for every ton of installed capacity still prevails.

E. M. MITTENDORFF, Charlottesville, Va. (WRITTEN): It appears to me that this paper must almost by necessity be read in conjunction with the same authors' paper entitled *A Proposed Procedure For The Design Of A Heat Pump Buried Coil* and issued in September 1954 by the Texas Engineering Experiment Station.

It is interesting to note how the increase of Reynolds Number did not change the rate of heat transfer.

Factor k in Equations 2 and 3, however, cannot be determined from available data, and together with my commending the authors for the splendid work done goes my suggestion that values for k and C_p are determined for various soils, so that the data in this paper may be put to practical use.

AUTHORS' CLOSURE (Mr. Fluker): Questions were raised in each of the discussions concerning the application of the coil length equation to a practical design problem and the lack of information available in the paper to make such an application. Clarification of these points will be considered first.

Application of the coil design equation, Equation 4 in the paper, it is agreed, will require a knowledge of the soil condition at the coil burial site plus practical methods of evaluating each of the quantities in the equation. Properties of the soil which are required and the proper methods to be used to evaluate each quantity for practical application of Equation 4 are given in *A Proposed Procedure for the Design of a Heat Pump Buried Coil*, by D. M. Vestal, Jr., and B. J. Fluker, Preliminary Copy, Texas Engineering Experiment Station, College Station, Texas, September 24, 1954. (Reference 1 in the paper.)

The proposed design procedure begins at the point where the current paper stopped. That is, the final expression for coil length, Equation 4 in the paper, which resulted from the dimensionless correlation of experimental data is assumed correct. The procedure presents step-by-step methods for determining conditions at the site. Quantities Q, Δt, k, and I are then computed on the basis of conditions found at the specific site, and Equation 4 is solved for coil length.

The coil design procedure was developed with the objective that the computed coil length be adequate to supply maximum loads during periods of extreme soil temperatures, soil moisture contents, and weather conditions. It is felt that this objective has been incorporated into the methods for evaluation of k, I, and t_r given in the proposed design procedure.

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The expression for k, Equation 3 in the paper, is given without derivation due to the space required to derive it satisfactorily. Suffice it to say, Equation 3 is empirical. It is to be the subject of a future paper which will present the soil heat transfer coefficient measurements work of this laboratory.

Mr. Mittendorff brings out an interesting point in his comments. He is correct in observing that virtually no change in rate of heat transfer may be noted when the circulating fluid Reynold's Number is changed. As discussed in the paper, the coil inside film coefficient plays a relatively minor role in the mechanism of heat transfer between fluid and soil. Similarly, the coil wall appears to have little effect upon the rate of heat flow. The properties of the soil immediately surrounding the coil and the coil outside film coefficient are the resistances limiting the flow of heat between fluid and soil.

In Fig. 3, as Mr. Borry has observed, there appears to be a number of distinct *chains* of data points forming comparatively smooth curves within the general scattering of points. Although not identified as such in Fig. 3, the data for each separate test actually plotted as a smooth *chain* of points with the ordinate decreasing asymtotically to a constant ordinate value as abscissa increased. The chain in the lower center of Fig. 3 represents a test which started normally with initially high values of ordinate and followed the general expected trend. At an abscissa of approximately 220, conditions were changed in this test without allowing an interval of time for the soil around the coil to regain its natural condition. Normally, tests were completed employing a constant set of fluid conditions through the duration of the test. It is interesting to note that this *chain* of points rapidly approaches the asymtote value of ordinate of the general data grouping.

The dimensional analysis study included 16 tests in which soil freezing to some extent was observed around the buried coils. Column (10) of Table 2 shows the maximum radius of soil freezing observed during these simulated evaporator runs. The effects of soil freezing in these cases was transient and appeared to have no noticeable effects upon the test result analyzed. For regions of the country with more severe climates than that found in Texas, the soil freezing problem may indeed become one of importance, as Mr. Borry states. Minimum soil temperatures in Texas at the 5-ft depth (test coil depth) were too high to permit soil freezing to any great extent around the coils with the refrigeration equipment employed.

The question posed by Mr. Borry concerning the coil operating temperature of a direct expansion system is one which we have not considered in the manner he suggests. We have considered the direct expansion system with respect to coil capacity no different from the indirect expansion system which we employed in our test set up. This is to say that, at thermal steady state, whatever the temperature conditions might be, the expression for coil length, Equation 4 of the paper, is thought to hold equally well for either a direct or an indirect expansion system. The critical point in designing a direct expansion system is in the selection of the coil design temperature. This temperature should be such that the coil capacity be minimum when the coil is operating at this temperature. Any other temperature conditions encountered may, as Mr. Borry suggests, produce a greater coil capacity which would be favorable to the overall system operation. In short, the coil should be designed to supply a specific load under the most adverse conditions anticipated.

The effects of increased coil running time upon the instantaneous rate of heat transfer between coil and soil may be realized by the following analysis.

If the coil is operating condenser say 50 percent of unit time (intermittency ratio = 0.50) the corresponding value of intermittency factor, obtained from Fig. 4a of the paper, is 2.3. If the operating time is increased to 100 percent the intermittency factor is reduced to 1.4. Substitution of these values for each case into Equation 2 of the paper and solving for Q, we have:

$$Q_{1.0} = (T_{1.0})(k\ L)(1.4)$$

and

$$Q_{0.5} = (T_{0.5})(k\ L)(2.3)$$

where the subscripts refer to the per cent running time. Then:

$$\frac{Q_{1.0}}{Q_{0.5}} = 0.61 \left(\frac{T_{1.0}}{T_{0.5}}\right)$$

if we assume L and k constant. Assuming $T_{1.0} = 1.25\ (T_{0.5})$ and solving for $Q_{1.0}$:

$$Q_{1.0} = 0.76\ (Q_{0.5})$$

This is to say that the instantaneous rate of heat transfer during continuous operation is 76 percent of the rate for half time operation where $T_{1.0} = 1.25\ (T_{0.5})$. Normally $T_{0.5} < T_{1.0} < 1.25\ T_{0.5}$ during condenser coil operation assuming other conditions identical.

The total capacity in any period of time for continuous coil operation is

$$C_{1.0} = 1.0(0.76\ Q_{0.5})(\theta)$$

where θ is time and C is total capacity. For half time operation

$$C_{0.5} = 0.5\ (Q_{0.5})(\theta)$$

Establishing the ratio

$$\frac{C_{1.0}}{C_{0.5}} = \frac{0.76\ (Q_{0.5})(\theta)}{0.5\ (Q_{0.5})(\theta)}$$

we obtain

$$C_{1.0} = 1.52\ C_{0.5}$$

We see, therefore, that the total capacity at continuous condenser coil operation is a maximum of 1.52 times greater than for half time operation under the conditions assumed.

A similar method of analysis may be employed to obtain these relationships for evaporator type coil operation.

1593

No. 1593

FROM '36 TO '56—AIR CONDITIONING COMES OF AGE

By WALTER A. GRANT*, SYRACUSE, N. Y.

DURING the first 35 years of this century, the air conditioning industry rose meteorically from a handful of ideas and a pocketful of capital to a place of prominence on the American business scene. It was an era of pioneering innovation, when much more was unknown than known, when imagination and insight produced leaping technical advances over the chasms in basic knowledge.

By contrast, the last 2 decades of air conditioning progress seem strangely quiet. This has been an era of improvement, of consolidation, and of acceptance. The industry has taken the brilliant technical and engineering advances prior to 1936, has translated them into eminently marketable products and systems, has produced and sold, and has installed and serviced them in volume. In 2 short decades it has achieved strength and maturity. Air conditioning has come of age.

One way to measure maturity and achievement is to review the hundreds of improvements, each minor in an individual sense, which have combined to produce the vastly superior products available today. The consumer is breathing cleaner air. His indoor environment is almost completely free of drafts. The air conditioning machinery is quieter, more foolproof, more durable, and costs less to maintain. For example, the average refrigerant charge in open compressor systems lasted about 3 years. Now, in a tight hermetic system it lasts indefinitely. A uniform indoor climate is now a fact all 12 months of the year. And all of this has been accomplished at progressively lower costs to the consumer.

One sure gage of an industry's maturity is the degree and nature of its acceptance. The steady upward growth of air conditioning sales to all markets is an accurate measure of the degree of acceptance. There are many insights into the changing nature of consumer acceptance. Only 20 years ago, air conditioning was a novelty. It was a heady adventure, a traumatic treat, to shiver in an air conditioned theater. The marquee outside publicized this fact with large hanging silver icicles. And it was invariably too cold and too damp inside.

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Presented at the 63rd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Chicago, February 1957.

Since then, people have learned to live with air conditioning. It has become an accepted fact of life. It is no longer necessary to freeze the consumer to prove concern about his welfare. People no longer operate either their heating or cooling equipment at 68 F. The spartan complex, both summer and winter, has given way to the temperature which the user prefers, instead of the temperature someone thinks he ought to enjoy. Gone is long underwear in winter and wilted collar and shirt in summer. Modern men and women wear clothing of about the same style and weight all year round.

The point is that the consuming public no longer thinks of air conditioning as providing an occasional cool thrill on a scorching day. It is now taken for granted, has become an integral part of everyday living. Air conditioning has come of age.

1936–40—THE PRE-WAR PERIOD

In 1936 the country was struggling to climb out of the trough of the great depression. Economic conditions throughout the thirties precluded any dramatic break-through in new markets although research continued under severe financial handicaps. During this period residential construction began to fall behind actual needs, to be stopped entirely during the war, and thus create the tremendous pent-up demand of the latter forties. Large office building construction had ceased. Building managements were more anxious to fill their echoing rooms with tenants than with conditioned air. Keen competition stimulated fitful activity in shops and other retail establishments, less than 5 percent of which had been air conditioned by 1936. Theaters which had pioneered comfort cooling through the twenties had reached about 15 percent saturation. And only about 1 percent of total factory floor space was air conditioned, concentrated in those industries where control of the environment was essential to production. Less than 1 percent of Class A office space was conditioned. There was no residential cooling business.

In view of its economic background, it is remarkable that the air conditioning industry itself was not smothered in its infancy. When it is considered that the industry had not yet developed a sound distribution network for its products and systems; that its financial condition left much to be desired; and that neither of its 2 major markets—residential and commercial buildings—had yet awakened, one is forced to conclude that the industry held forth an economic need that would not be denied. Clearly distribution, sales promotion, and consumer acceptance would have to be stepped up to permit volume production to keep pace with the industry's record of technical progress.

Distribution in the early days was entirely through the contracting of specially engineered and tailored installations. In many of the larger cities, there were capable consulting engineers who were able to design plants for commercial customers, for bid and installation by mechanical contractors. But in large areas of the country, and for industrial applications generally, the client still looked to the experienced air conditioning contractor to perform the engineering and installation services he required.

Yet in this pre-war period the basic steps had been taken toward industrial strength through wider distribution. In the fall of 1932 the first dealer organization had been created by Carrier to be followed within a few years by other companies. Early dealerships were operated by a conglomeration of entrepreneurs—heating and ventilating contractors, commercial refrigeration fixture outlets, plumbing houses, electrical contractors, building supply jobbers, hardware stores—in fact

anyone with a little capital and a large faith in air conditioning's future. The major contribution made by the early dealer was financial, since he knew little about designing, selling, installing, and servicing. The manufacturer continued to perform these functions for some time while he trained the dealer in all the specialized operations of the business. There was virtually no distributor-dealer relationship at this time, since volume distribution had not begun. Because practically all installations had to be engineered into the customer's building, sales people of the period were preponderantly engineers. In 1936 the dealer segment of the industry was still wrestling with air conditioning's technical complexities, but it learned fast, and by 1940 dealers were handling around 30 percent of the business.

Amid all of these problems of distribution, important product advances were achieved. The first self-contained units for the commercial market appeared, and room air conditioners began their bid for public favor. Highly significant improvements were made in refrigerating compressors and heat exchangers, stimulated by the new fluorinated refrigerants. The future looked bright, and more and more companies started to enter the air conditioning business through formation, merger, or acquisition.

1941–45—WORLD WAR II

In the early war days, business leaders believed that wartime needs for the industry's products would be so minor that wholesale conversion to direct military production would be required. This did not prove to be the case. About 80 percent of the industry's manufactured output consisted of normal civilian or special military designs of air conditioning products and systems. But they were entirely for military or essential civilian end use. The industry operated under many handicaps, and material shortages became a severe problem. Successive scarcity of nickel, chrome and copper led to substitutions of steel and plastics in many products and components, and taxed the ingenuity of design and application engineers.

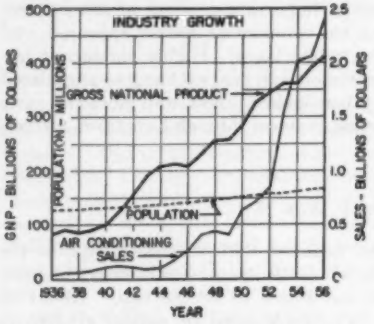
Process air conditioning was used in numerous war-related industries—for precision machining, and instrument and communications equipment manufacturing, to hold tolerances and protect against corrosion and grit; for pharmaceuticals, to provide uniformity, control chemical reactions, and inhibit bacteria; for bombsights, telescopes, and other optical equipment, to control dust and dimensions; for powder and explosives, to retard deterioration and assure accurate timing; and so on, through electronic tubes, textile fibers, plastics, and the processing, storage and transportation of food.

Beyond essential industrial uses were the direct military requirements. For the Air Force were control towers, celestial navigation and Link trainer rooms, storage and maintenance of bombsights, parachutes and flight clothing, engine test buildings, and photographic trailers. For the Army were hospitals, commissaries, rooms for instrument and equipment repair and adjustment, and mobile units of many types. For the Navy, air conditioning went to sea in submarines, battleships, cruisers, destroyers and aircraft carriers.

Possibly 20 percent of the industry's manufacturing facilities were devoted to direct war work unrelated to air conditioning. Tank gun mounts, magnetic brakes, searchlight parts, anti-submarine devices, air-frame components, and shell loading formed but a small part of this activity. Air conditioning manufacturers conducted service and maintenance schools for the armed forces and performed a substantial

amount of research and development work for the government, much of it at the manufacturers' expense. Representatives of the industry served on the Industry Advisory Committee to the War Production Board and in many other important capacities in government service.

These were years of peak output during which major research energy was expended on special applications. Advances in products and techniques were directly related to the major problem at hand—winning the war. Air cycle cooling was developed for aircraft, and dehumidifying equipment now used in damp basements was designed for subterranean bomb shelters. The massive low temperature air



Air conditioning sales are at consumer level, installed. Excluded are domestic and commercial refrigeration equipment, heating, and cooling towers. Statistics in Figs. 1–5 represent best available judgment applied to data from U. S. Department of Commerce, *ACRMA*, *ARI*, and other reliable sources, with estimates made for years lacking adequate data.

FIG. 1—GROWTH OF AIR CONDITIONING SALES IN THE LAST 20 YEARS IN RELATION TO THE GROWTH OF POPULATION AND THE OVERALL ECONOMY

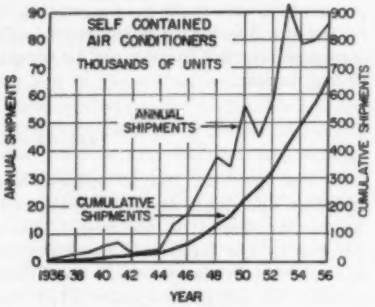


FIG. 2—NUMBER OF SELF-CONTAINED AIR CONDITIONERS SHIPPED IN THE PERIOD 1936 TO 1956. NOTE THE SHARP TRANSITION AT THE END OF WORLD WAR II FROM THE PIONEERING TO THE ACCEPTANCE STAGE OF THE BUSINESS

conditioning plant for the Cleveland N. A. C. A. wind tunnel and York's all-weather stratosphere chamber for personnel testing are merely 2 of many outstanding applications. While new civilian comfort air conditioning was nonexistent, the experience of thousands of service and civilian personnel with air conditioned structures like the Pentagon in Washington and blackout plants throughout the land awakened a desire for air conditioning and set the stage for the market expansion which was to follow.

The war proved 2 points: that air conditioning was essential to the war effort; and that the men and women of the industry had accomplished miracles in supporting the military program. The country began to realize, indeed, that air conditioning was coming of age.

1946–56—THE POST-WAR PERIOD

The industry emerged from the conflict with relatively few conversion problems and set about effecting its transition to maturity. Prior to the war, it had relied

upon engineering and application skill to pull it through. Post-war, it exploited all of its resources. The merchandising and marketing arm of the business was greatly strengthened by the development of self-contained equipment. Advertising, largely restricted in the thirties to trade magazines, has since been employing and exploiting every available medium.

Fig. 1 illustrates in broad terms the success achieved in this post-war decade. While population between 1936 and 1956 increased by less than a third, and gross

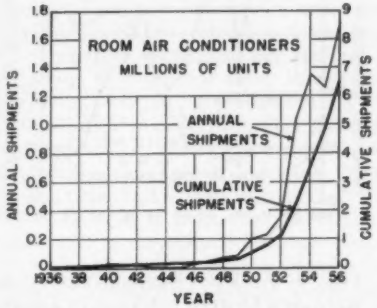


FIG. 3—NUMBER OF ROOM AIR CONDITIONERS SHIPPED IN THE LAST 20 YEARS. NOTE THAT IN THE LAST 4 YEARS, MORE THAN 5 TIMES AS MANY UNITS WERE SHIPPED AS IN THE ENTIRE 16 YEARS PREVIOUS

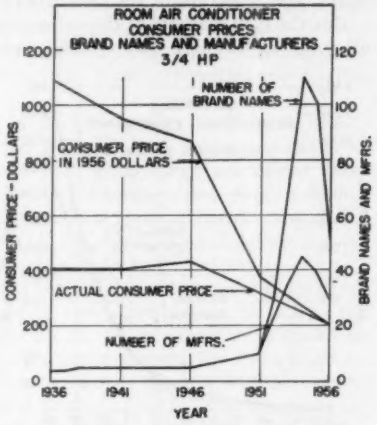


FIG. 4—NUMBER OF ROOM AIR CONDITIONER BRAND NAMES AND MANUFACTURERS REACHED AN ALL TIME HIGH IN 1954. DUE TO SALES VOLUME AND PRODUCT IMPROVEMENT IN A FREE COMPETITIVE ECONOMY, CONSUMER PRICES FELL DRAMATICALLY FROM 1936 TO 1956

national product increased 5 times, the value of air conditioning sales at consumer prices soared 50-fold from over $48,000,000 to $2,400,000,000. And most of this increase has occurred since 1946. This curve and the post-war sales curves for typical air conditioning products (Figs. 2, 3 and 5) bear a striking similarity, with first a gradual rise, and then a breakaway, illustrating a typical growth pattern. During the thirties air conditioning was completing its apprenticeship to the industrial roster, a period of great innovation and trial. Since the war the era of public acceptance has dawned, and the entry of many new companies (Fig. 4) showed that this period was here to stay.

But Figs. 3 and 4 illustrating the room air conditioner picture demonstrate far more. In 1936 only about 3,000 units were sold compared with 1,700,000 in 1956. During that interval the retail price has been cut in half from $400 to around $200. But the 1936 price expressed in 1956 dollars is $1,100 which is 5-½ times the price it can be purchased for today. Since the war, the manufacturing industry has

made an enormous investment in new plant and production tools. Here truly is an example of the American competitive system at work to bring better products and values to the consumer.

Yet the principal air conditioning markets are still far from saturated. In 1936, there were only 2 markets where more than 1 percent of the establishments were cooled. This picture has changed tremendously in the past 20 years. It is now estimated that about 90 percent of all theaters, 35 percent of motels, 27 percent of Class A commercial office buildings, 15 percent of hotel guest rooms and retail stores, and 3 percent of the churches in this country are air conditioned.

But the lush markets of the future are still practically untouched. Considering all types of dwelling units—single and multi-family houses, apartments, mobile

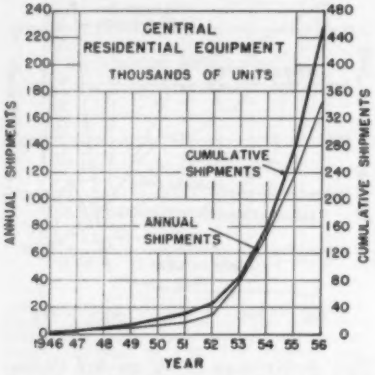


FIG. 5—CENTRAL RESIDENTIAL EQUIPMENT WAS NON-EXISTENT 10 YEARS AGO, IS ONLY NOW REACHING ITS STRIDE

FIG. 6—ASHAE EXPENDITURES FOR RESEARCH DURING THE LAST 20 YEARS. NOTE THE SHARPLY DECREASING RATE WHEN MEASURED AS PERCENT OF AIR CONDITIONING SALES CHARTED IN FIG. 1

homes—only about 6 percent have any form of cooling at all. Although annual shipments of central cooling units in 1956 were around 175,000 units (Fig. 5), the total installed to date represents only between 1 and 2 percent of saturation. And the other great potential market, comfort cooling of factories, is perhaps 2 percent air conditioned today.

The post-war era has also seen the rise of markets barely touched in the thirties. Automobile air conditioning was not a factor until 1951 when about 2,000 units were installed. The 1956 estimate is 300,000 units, and sales for 1961 are projected at 2 million. Another new market is the 600,000 existing and 110,000 annual construction of house trailers, where air conditioning is proving just as attractive as in fixed residences.

The tremendous growth in the last 20 years would not have been possible without a solid foundation of scientific and engineering achievement. Let us review the record in detail.

BASIC KNOWLEDGE

The foundation for all technological advance is expansion of basic knowledge. Such an expansion has occurred in large measure during the past 20 years. It is

impossible in a paper of this scope to dwell upon the advances in all the areas of science which have influenced the growth of air conditioning. Of interest here is the progress in fundamental knowledge in those particular areas of physics, thermodynamics, heat transfer, fluid flow, and physiology which have had direct impact upon air conditioning.

To present a bibliography would require a volume in itself. Fortunately, the important references are listed in the current ASHAE HEATING VENTILATING AIR CONDITIONING GUIDE, the ASHAE TRANSACTIONS, the *ASRE* Data Books, and in Refrigeration Abstracts. The reader is referred to these sources for details about accomplishments and the people who have been responsible for them.

Our Society has conducted and sponsored much important basic research since establishment of its research program in 1919. Fig. 6 shows how this activity has grown since 1936.

In the very forefront has been research in the physiological and medical field. The effect of thermal shock has been studied and evaluated, and much has been learned about the influence of both extreme and optimum air conditions upon healthy people, on persons afflicted with organic disorders, and those suffering from respiratory infections and allergies. Gradually a better understanding of the sensation of comfort is being achieved, and of the effect of air conditioning on human productivity and efficiency. But much remains to be learned.

Basic work on odors is now being conducted both in this country and in England by chemists, physicists, physiologists, biologists, and psychologists, as well as by engineers. The Technical Advisory Committee on Odors assembled authorities in their respective fields at the Basic Odor Research Correlation Conference in New York City in 1953. Further fundamental studies at the ASHAE Laboratory resulted in a very important paper presented at the ASHAE annual meeting in 1955. As soon as the basic mechanism of odor perception is understood it will be possible to control odors in air-conditioned spaces.

In the field of thermodynamics, important additional data have been obtained on the properties of moist air, leading to improved psychrometric charts and tables. Results obtained during the past 10 years from investigations into the nature of vapor transmission through materials represent a real advance toward developing practical methods for preventing unwanted condensation. Absorption and adsorption phenomena have been further explored. More accurate values of film coefficients for gases, liquids, and boiling and condensing vapors have been obtained, as well as certain basic properties of materials, such as conductivity, specific heat, density, and viscosity. Some knowledge has been gained about the mechanism of frost formation.

Advances in heat transfer have been many and important. Periodic and transient heat flow analyses and experiments have led to more accurate evaluations of cooling loads in structures. An important step in facilitating analyses was the development of the sol-air temperature concept by Mackey, and an increasingly useful tool has been the application of analogue computers for solving heat flow problems. (Fig. 7).

Recognition that the heat storage of building materials may be utilized to level out the calculated instantaneous cooling load has significantly reduced the size of equipment installed in residences and commercial buildings. A by-product of smaller capacities has been better control of humidity and improved comfort conditions. Of substantial importance to the application of air conditioning systems has been increased knowledge in control system theory and engineering. The

thermal storage effect of the radiant portions of solar and lighting loads is better known. Research activity along this line has been accelerating, and a better understanding of how and when an impressed heat gain becomes equipment load is imminent.

Research into properties of glass and plastics, and their influence on solar heat gains, has prompted the development of heat absorbing glasses and improved

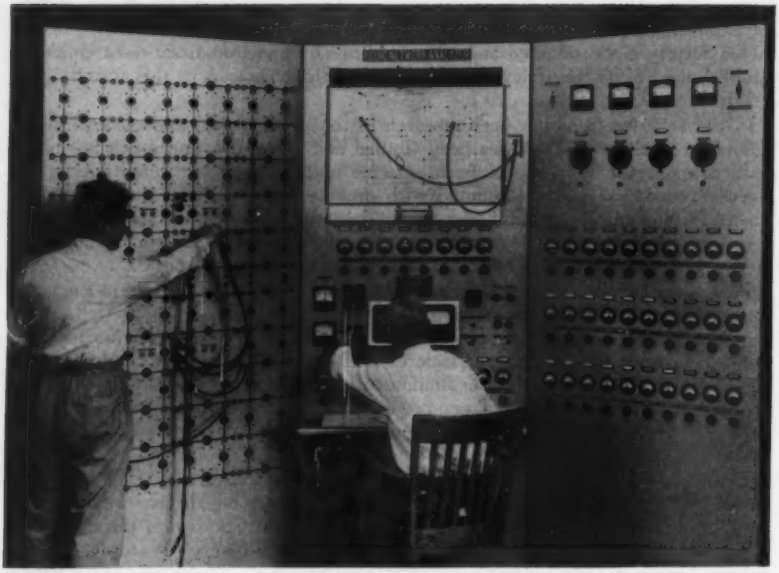


FIG. 7—TYPICAL ELECTRIC ANALOGUE COMPUTER IN OPERATION. COMPUTERS SIMILAR TO THIS ARE PROVING VERY USEFUL IN ANALYZING COMPLEX THERMAL CIRCUIT PROBLEMS RELATING TO HEATING AND COOLING LOADS IN BUILDINGS

shading devices. In the realm of heating an outstanding area of progress is the entire theory of panel heating, including data on the radiant and convective portions of the useful heat output under a variety of configurations and conditions. Important for the heat pump, has come an extensive examination of heat sources, including air, water, ground, and solar radiation.

In fluid dynamics, important additions have been made to fundamental knowledge of compressible flow, which have led to design improvements in turbo-machinery. Incompressible flow has also received attention: examples include the improved friction chart for air flow in straight pipe; more information about the mechanism of energy losses in duct fittings; and refinements in the static regain concept. Some advances have been made in the flow of air within conditioned spaces, involving the complex factors of induction, turbulence, mixing, heat transfer,

buoyancy, and inertia, but much remains to be done. The surface of the problem has hardly been scratched.

The rapid strides made in atomic physics during the past 20 years in knowledge of the structure of matter are bound to affect air conditioning. Semi-conductors have been revolutionizing electronics and research work is in progress to use electricity directly to produce cooling through the Peltier effect. Research has accumulated more basic information on sound and vibration-generation, transmission, isolation, and absorption. With depletion of natural resources of fossil fuels, utilization

FIG. 8—RESEARCH WORK IN PROCESS AT THE ASHAE LABORATORY IN CLEVELAND. PICTURED IS A TEST UNIT TO MEASURE THE BURNING CHARACTERISTICS OF FUEL OILS

of solar energy for heating or cooling has been receiving increased attention. Better data are available on its geographical availability, and improvements have been made in collecting and concentrating devices.

AIR CONDITIONING EQUIPMENT

In 1936, Willis H. Carrier in his paper†, described in detail the spray type air conditioner first designed and built in 1903, and developed to a high efficiency in the years to follow. He also noted the first use of cooling coils, and the development of unitary air conditioners, both fan-coil and completely self-contained.

The double decade since his review has been marked by the almost complete swing to coil-type equipment, relegating the spray air conditioner to industrial applications where it still has a special advantage. The number of types and arrangements of fan-coil and self-contained units is beyond count, and is a consequence of the fantastically accelerating demand for products which will meet every conceivable application requirement.

In the late thirties, the self-contained summer air conditioner, which included all the elements of the refrigeration cycle, moved in 2 directions: (*a*) equipment for

† Progress in Air Conditioning in the Last Quarter Century, by W. H. Carrier (ASHVE TRANSACTIONS, Vol. 42, 1936, p. 321).

cooling small rooms and (*b*) equipment for cooling larger spaces. The former, optimistically called a portable room cooler at the time, has since been officially named the room air conditioner for sizes below 1-½ hp. The category above 2 hp, now called self-contained units, is subdivided into styled equipment for use within the space, and equipment for remote location employing fans suitable for duct-work.

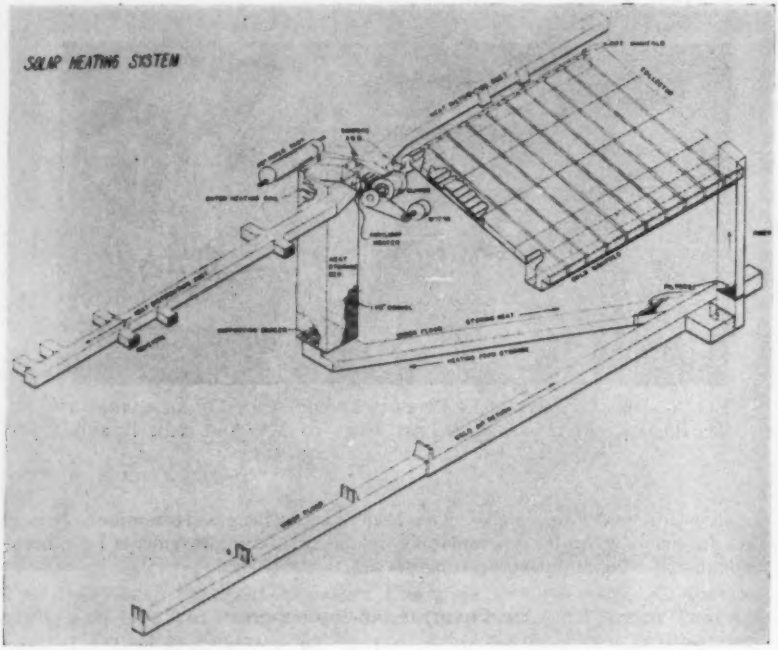


FIG. 9—SOLAR HEATING SYSTEM FOR THE DENVER DESIGN HOUSE

Room Air Conditioners: Early room air conditioners which employed water-cooled condensers and required plumbing and drain connections did not lend themselves well to appliance selling. The air-cooled model was welcomed because of its easy application and suitability to mass production and market techniques. It is the first product the industry has developed which can be classed as a major household appliance. Much work has gone into perfecting it in the last 20 years. Noteworthy have been the high-speed hermetic direct-drive compressor; the capillary feed device; quieter fans; increased use of plastics; improved compressor and fan motors; and substitution of aluminum for copper in heat transfer components. These developments have brought much better space utilization. In 1936, the volume occupied by a ¾ hp unit was over 15 cu ft; by 1956 the average was down

to about 7 cu ft. The same so-called portable unit weighed 465 lb in 1936; the comparable 1956 figure is 200 lb.

Early room air conditioners were all consoles. The first window sill model made its bow in the late thirties and by 1950 its lower price had practically squeezed the console out of the picture.

In the last 2 or 3 years, sales volume has been large enough to justify diversified model types to satisfy particular application requirements. Window units are

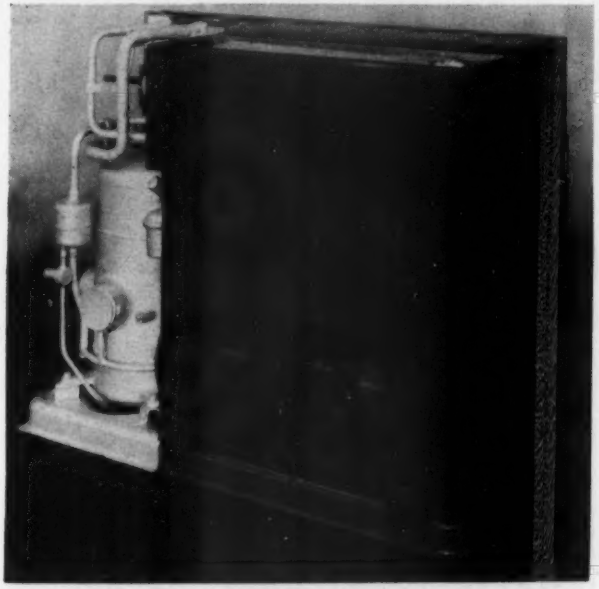


FIG. 10—THIS CONSOLE ROOM AIR CONDITIONER (CASING REMOVED) OF VINTAGE 1933–37 WAS 20 YEARS AHEAD OF ITS TIME, FEATURED A HERMETIC COMPRESSOR AND REVERSE CYCLE HEATING, AND WON FOR ITS DEVELOPMENT ENGINEERS HENRY L. GALSON, HENRY C. HELLER, CHARLES R. NEESON, AND HAWS K. STEINFELD THE JOHN SCOTT MEDAL AWARDED BY THE CITY OF PHILADELPHIA

showing a definite trend toward a thinner silhouette with minimum overhang indoors and out. They are also receiving increasing attention for through-the-wall application, signalling their acceptance for permanent instead of makeshift installations. Recent Underwriters' rulings have made advantageous the development of 7-½ and 12 ampere units to permit plug-in on conventional house wiring circuits. Consoles are again being revived. Since they are completely inside the room and obstruct minimum light, they offer a positive advantage over the window unit. Because all the functions of a heat pump are present in an air-cooled room air conditioner, this feature, which was unsuccessfully offered in the early thirties, is again available as an optional extra.

Self-Contained Units: The self-contained unit has changed little during its 20-year existence. It was conceived for smaller stores and restaurants, which had set the pace on commercial air conditioning. The requirements were simple—minimum floor space, unobstructed air flow, and styling suitable for location within the room.

Today's self-contained unit looks much like its 1936 predecessor. Internal improvements have been similar to room air conditioners, including hermetic com-

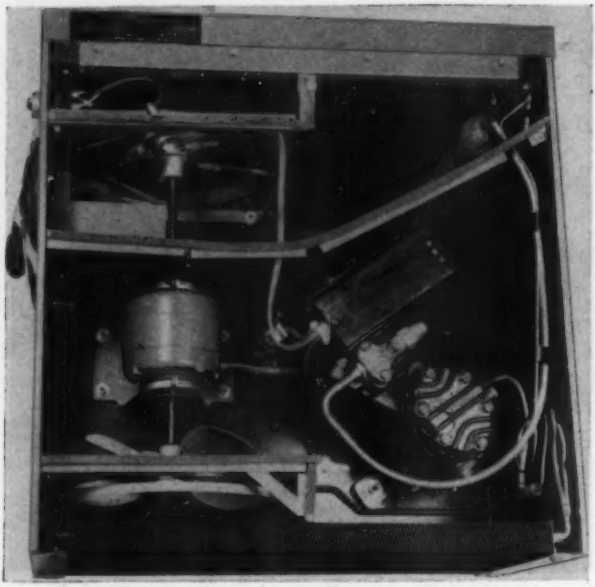


FIG. 11—TOP VIEW OF MODERN WINDOW MODEL ROOM AIR CONDITIONER, SHOWING EVAPORATOR FAN AND COIL (TOP), CONDENSER FAN AND COIL (BOTTOM), AND HERMETIC COMPRESSOR (RIGHT)

pressors, a trend toward capillary expansion devices, and improved electric motors and control. Recently there has been more emphasis on styling. In the early thirties, casings were primarily functional, whereas today, units up to about 7-½ hp are styled to blend inconspicuously with most kinds of decor.

Until recently all self-contained units were water-cooled, using city water to waste or a cooling tower. Within the past 5 years there has been a trend toward air-cooled models. Since a one-piece air-cooled package is bulky and poorly suited to the small store which does not have ready access to outdoor condenser air, most air-cooled models are 2-piece. The condenser is remotely located, and the 2 packages are interconnected by refrigerant piping.

In the post-war period there has been the start of a trend toward ceiling suspended units, paced by the needs of chain stores where profit margins require every

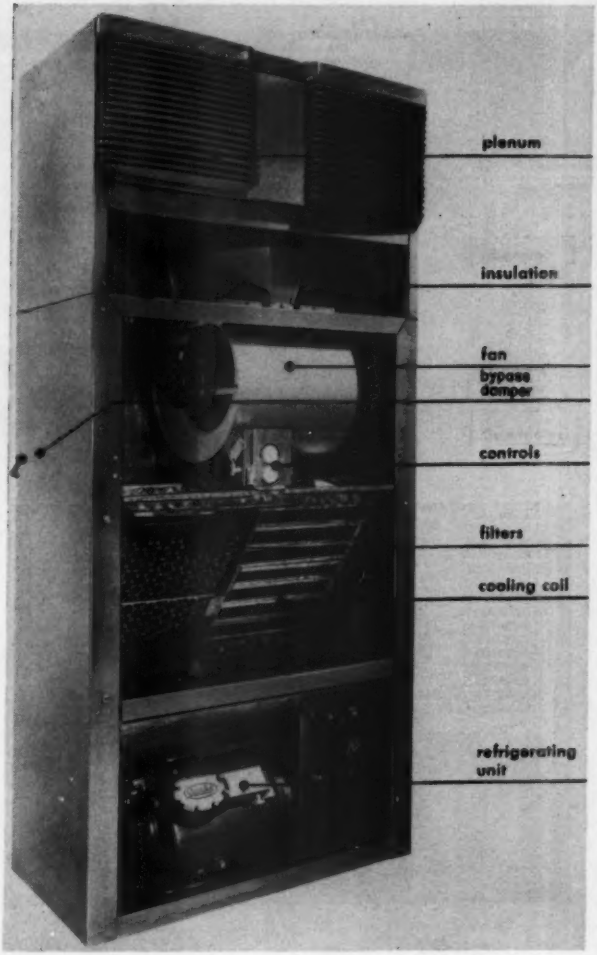


FIG. 12—1956 MODEL 5-TON SELF-CONTAINED AIR CONDITIONING UNIT FOR COMMERCIAL ESTABLISHMENTS, SHOWING PRINCIPAL FUNCTIONAL ELEMENTS

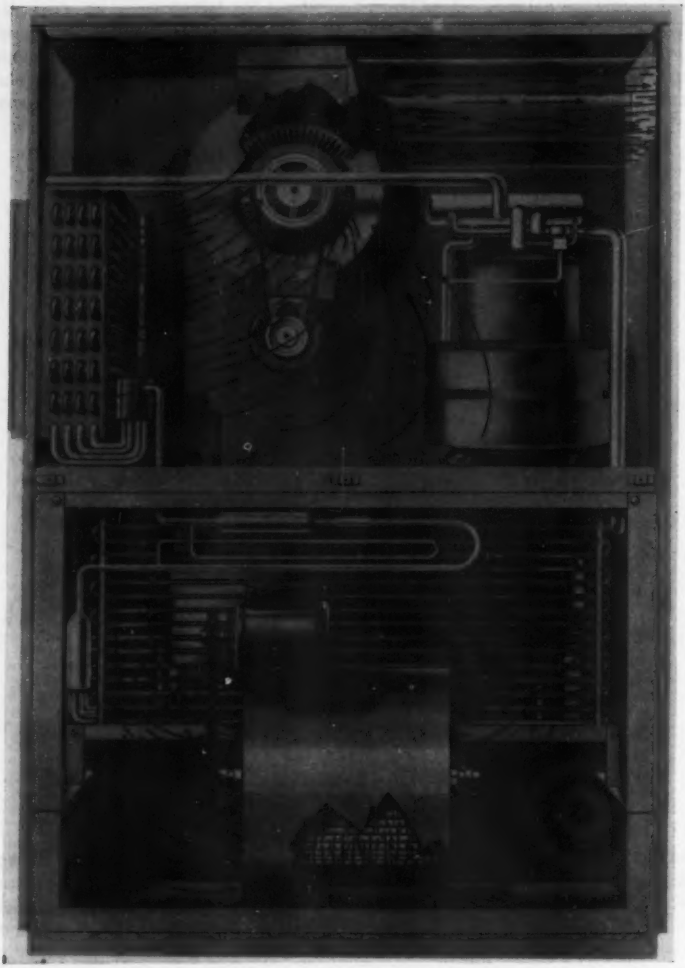


FIG. 13—SINGLE PACKAGE AIR-TO-AIR HEAT PUMP UNIT FOR HOMES AND COMMERCIAL ESTABLISHMENTS

possible square foot of floor area for merchandise. More recently the range of sizes has been expanded up through 15 tons for equipment located within the room, and considerably larger for units located outside the room. This trend has been made possible by increased manufacturing volume, which has reduced costs below field-assembled components. This has resulted in cost savings to the consumer, who today finds scant justification for assembling components below 25 tons in one unit. In fact, the economics often permit large installations to be best served by a multiplicity of small self-contained units.

Heat Pump: The specialized self-contained unit called the packaged heat pump is a phenomenon of the last 20 years. Early installations, like General Electric's first heat pump in 1935, were assembled systems using fan coil units or central system equipment and remote compressors.

Considering how long the basic knowledge and design requirements have been known and recognized, heat pumps have made rather slow progress. By 1944 less than 100 commercial and residential units had been installed. By 1950 they numbered less than 1,000, and every year since, rosy predictions have outrun actual sales.

The organization which has done most to promote the heat pump is the American Gas & Electric Service Corp. which in the early thirties recognized the basic economic advantages to the electric utility industry, pioneered many early installations, and has consistently promoted the idea. The relative slowness of the industry to take the ball and run with it has been due primarily to economic factors. Since the heat pump is a full-fledged cooling machine, the buyer has to be a prospect for summer cooling before he can justify a heat pump. But it is only during the last 5 years that the market for residential cooling has even begun to break.

The heat pump costs more than conventional equipment. Except in a few favored locations, operating cost during the heating season is substantially more than with conventional fuel. Cost reduction by mass production, as well as considerable sales promotion, will be required to overcome these elements of resistance.

During the past 7 or 8 years, water-cooled self-contained units have frequently been converted to water source heat pumps, and this has become a job shop business in certain parts of the country. But the water source pump is advantageous only where there is a cheap and plentiful source of well water, river, or lake water. While there are many such locations, they represent a small proportion of air conditioning prospects, and the market thus far has been confined to a relatively small area in the south.

The air source pump has not proved to be a simple conversion since the technical problems have been more difficult. Because available heat declines with decreasing outdoor temperature, while the heating requirement increases, the deficiency must be made up by supplemental heat; otherwise the heat pump becomes excessively large. The recent trend has been toward use of electric resistance heaters to supplement the heat pump at low outdoor temperatures. Much development work is now in process to overcome the outdoor coil frosting problem and to simplify the switch-over valve and control.

Two styles of air source heat pumps are currently offered. The one-piece heat pump combines all elements into a single but rather large package, thereby minimizing installation labor. However, it requires a location which has access to outside air for heat rejection. The first production air source unitary pump was introduced by Drayer-Hanson in 1945.

The 2-piece heat pump comprises one package located outdoors or adjacent

thereto for absorbing and dissipating heat, and an indoor package to heat and cool the house. The 2 packages are interconnected with refrigerating piping.

Fan Coil Units: Nothing revolutionary has occurred in the design of fan coil units during the past 20 years. Their pattern was established in the early thirties

FIG. 14—TYPICAL GRAVITY WARM-AIR FURNACE OF THE MIDDLE THIRTIES, FAMILIARLY KNOWN AS THE OCTOPUS

in both the small room unit and the larger remotely located unit. In both categories there has been an increase in varieties and sizes available, better to meet the many application requirements of the remote systems they serve. One notable improvement is the *zoning type* fan coil unit with a multiplicity of dampered supply connections to serve a corresponding number of zoned spaces. Other improvements in fan coil equipment parallel those in self-contained units.

Small and large fan coil units, which are bread and butter business to the industry, have proved very advantageous in large commercial and industrial buildings where a central source of chilled water is used. There is also a market for room type units in residences.

HEATING EQUIPMENT

While the purpose of this paper is to review progress in air conditioning, the advances made in heating which is indispensable to the basic requirement of temperature control are also properly a subject for appraisal. For, despite the ripe

old age of the heating business, there have been very important and significant improvements in products and systems during the past 20 years.

Homes, commercial and industrial buildings are much more comfortable and livable in the winter of 1956 than they were in 1936. While tremendous credit

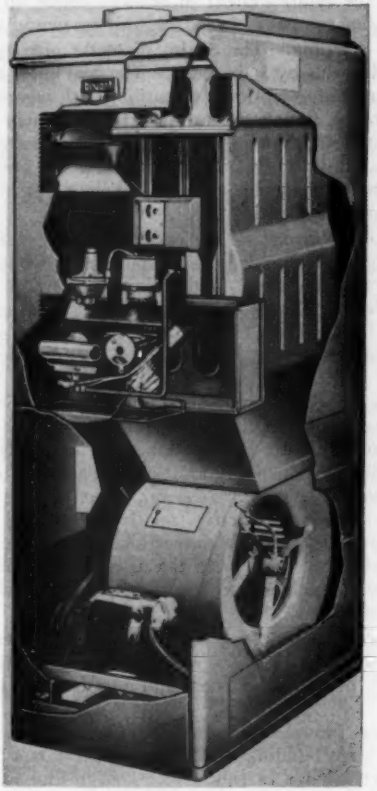


FIG. 15—MODERN GAS-FIRED FORCED WARM AIR HEATING UNIT FOR HOMES

should go to the advances in heating technology recited below, one should not lose sight of the fact that basic improvements have been made in sealing buildings much more effectively from wind and weather through insulation, weather-stripping, moisture barriers, and double glazing. These techniques have complemented the advances in heating, and have greatly simplified the task of the heating engineer.

The developments which overshadow all others in the heating field are the dramatic trends from coal to oil and gas, and from manual to automatic firing.

Residential Heating: In 1936 only 30 percent of new heating equipment sold was automatic. Today, the figure is closer to 99 percent, with about 65 percent of the installations gas and 34 percent oil. In 1936 most equipment was designed for coal firing and then modified to use gas or oil, although a few companies pioneered in designs specifically for these fuels. With the northward and eastward expansion of gas pipelines and the establishment of broader crude oil distribution, many equipment manufacturing companies in the mid-thirties swung onto the band wagon.

Results have been startling. The 54 in. diameter gravity *octopus* has given way to equipment 24 in. wide by 34 in. deep. The coal bin is now the hobby shop. Miniaturization has created exceptional versatility with regard to location. Slender, vertical or horizontal, upflow or downflow furnaces may be placed in utility closets, crawl spaces, attics, or garages, as well as in the basement. Much greater emphasis has been placed on styling since the furnace has been liberated from the basement to the living areas of the home. Judicious use of color is seen, with pleasing lines and general aesthetic appeal.

Twenty years ago most furnaces were cast iron. The shift has been strongly toward steel as welding techniques and the cost picture have improved. The typical 1936 furnace had a radiation shield for its major insulation. Today, glass fiber materials and reflective metal surfaces are performing more efficiently and at less cost to reduce casing heat losses. Heat exchangers are more efficient, and ceramic coatings or aluminized or stainless steel provide added protection from high temperature corrosion.

The large cast-iron boiler has been scaled down to a fraction of its former size. For residential heating, steam has given ground to hot water, and gravity hot water has been largely replaced by forced circulation. A pump in the water line permits installation of the boiler in almost any location and achieves more rapid and even heating response. Another sign of progress in the past 20 years has been the retirement of the old-fashioned, unattractive, poorly performing radiator. Distribution of heat is now accomplished by convectors or baseboard which perform inconspicuously and better. Attention to the design and location of these devices has resulted in a reduction of floor-to-ceiling temperature stratification, thus greatly improving comfort conditions.

Much has been learned since 1936 about conditions necessary for comfort in residences. Within the past 10 years the industry has developed perimeter loop systems which largely prevent cold floors and appear to be gaining favorable acceptance.

The past decade has brought an unusual amount of activity in panel heating, both for homes and large buildings. ASHAE research has been in the forefront in advancing basic knowledge in this field. Design and installation techniques have been brought to a high degree of perfection for floor, ceiling, and wall panels. The most popular form in residences has been the heated slab, installed directly on the ground with imbedded steel or copper pipes, edge-insulated to prevent undue heat loss. The control problem has been difficult, due to thermal lag and capacity, but progress has been made in improving control systems for this duty. Considerable work has also been done on ceiling panels with pipe or tubing imbedded in the plaster.

Many of these advances are a direct result of better cooperation among manufacturers through industry research, and industry- and association-sponsored university research. The publication in manual form of research results and the increasing use of industry standards has been of material benefit to all parties.

A phenomenon arising in the wake of cheap public power has been direct electric heating of homes; and there are thousands of such homes thus equipped in the TVA areas and in the Pacific northwest.

Commercial and Industrial Heating: Commercial heating has developed along lines similar to residential heating. Equipment dimensions have been reduced, and a greater variety of types and sizes is being offered. The most pronounced trend has been the use of factory-assembled units to reduce the labor of field-assembly and installation.

New industrial building concepts have had an important effect on heating techniques used in industrial enclosures. The broad expanses of the modern one-

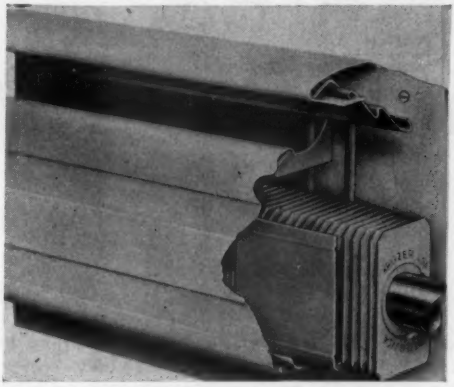


FIG. 16—UNOBTRUSIVE RADIANT AND CONVECTIVE BASEBOARD HEATING HAS SUPERSEDED THE UGLY RADIATOR, AND GIVES SUPERIOR RESULTS

story building have posed both heating and ventilation problems not present in multi-story factories, particularly in the windowless central sections of these one-story plants. In consequence, the old heat diffuser has evolved into a heating and ventilating unit.

Development of new combustion chamber materials, stainless, aluminum-coated, and ceramic-coated steel, has made it practical to develop automatic self-contained direct-fired heating systems of all sizes, with significant savings in first cost. A major impetus to the use of direct-fired industrial equipment has been the tremendous increase in the use of fuel oil and natural gas.

This trend has carried over into boiler plants. Built-up boiler plants with field-installed fire brick and structure have given way to all-welded packaged boiler-burner units (Fig. 17) delivered to the site complete and ready to fire with merely the addition of fuel, power. and flue.

AIR PURIFICATION, MOVING, AND DISTRIBUTION EQUIPMENT

Air Cleaning: By the mid-thirties the industry had come a long way toward adequate air cleansing. Early filtration methods included an array of spray nozzles

blowing across the stream of air. By varying the temperature of the water it became possible to obtain good humidity control, but the method was not effective in separating smudging dirt from the air.

The bag-type strainer and the dry filter next evolved were more efficient but offered substantial resistance to air flow. Both cleanable and automatic viscous filters developed during this period using the impingement principle proved to be effective on large dirt particles, less effective on fine dirt, and ineffective on smoke.

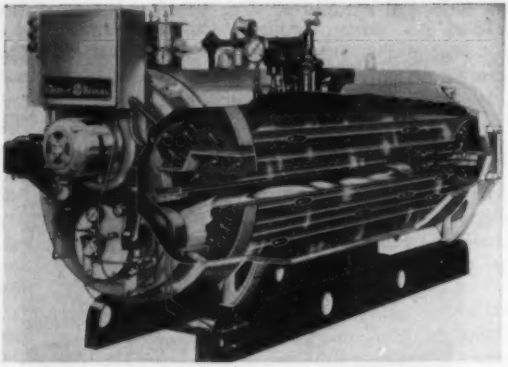


FIG. 17—THE AUTOMATIC GAS- OR OIL-FIRED PACKAGED STEAM GENERATOR HAS GREATLY SIMPLIFIED BOILER INSTALLATIONS FOR MEDIUM CAPACITY HEATING OF COMMERCIAL AND INDUSTRIAL BUILDINGS

Inexpensive throwaway filters appeared on the market which eliminated the necessity for frequent washing and re-oiling.

In the middle thirties, Westinghouse developed the electrostatic cleaner, a modification of the Cottrell precipitator, which removes all sizes of dirt, including smoke. Electrostatic precipitation was also introduced in combination with dry filter media.

Dry filters have been recently developed which strain out materials as fine as smoke and radioactive fallout. These offer higher resistance to air flow than the electrostatic, but are less expensive.

There has been a gradual acceptance that high quality cleaning reduces damage to goods from soiling and costs of washing, dry cleaning and redecorating due to smudging, caused by the carbonaceous particles in smoke and soot floating in industrial communities.

A large percentage of the population suffers from some form of nasal allergy, usually ragweed pollen or house dust. Here again, effective removal techniques have been developed in the last 20 years which relieve the condition in practically all cases while the occupant is indoors.

Odor Control: As stated under Basic Knowledge, much work in the realm of basic science is still required on the physiology of odors. However, the application techniques of using activated charcoal to absorb odors from the air have been greatly

improved in the past 15 years, thanks primarily to pioneering by the Connor Engineering Corp. Most of the mystery has been taken out of odor control by putting it on an economic basis, whereby capital and operating costs of heating and cooling equipment are reduced through the reprocessing of recirculated air with a consequent reduction in the amount of ventilation air taken into the building.

Particularly during the past decade there has been tremendous promotion of volatile additive perfumes, many of them advertising *activated chlorophyll* and other

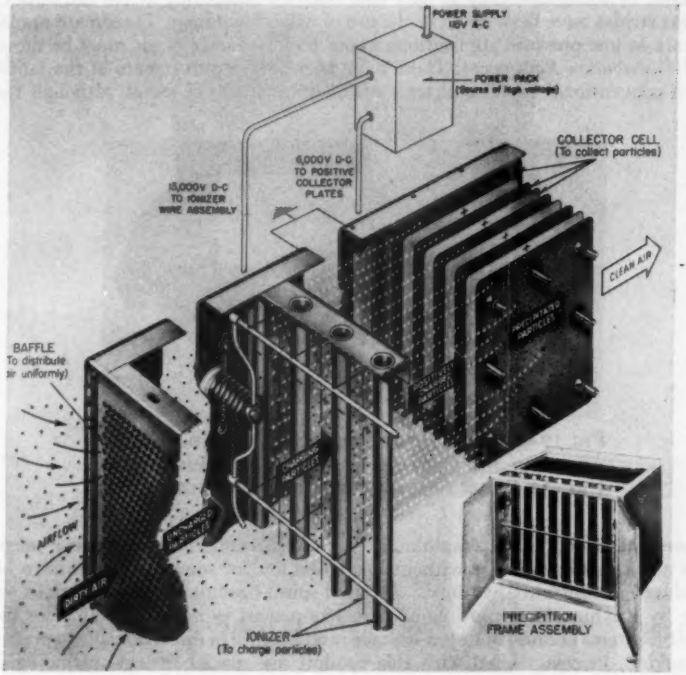


FIG. 18—THE ELECTRIC PRECIPITATOR WAS THE FIRST AIR CLEANER TO REMOVE EFFICIENTLY FINE DUST, SOOT AND SMOKE FROM ATMOSPHERIC AIR USED FOR THE VENTILATION OF BUILDINGS

reminders of outdoor freshness. The precise action of these materials on odors has not been definitely established. While their chief utility is to chase away kitchen and bathroom odors in the home, there has been considerable use for them in air conditioning systems. In some instances there has been user satisfaction, in others the practice has been abandoned after a year or so of trial.

Fans: By the start of the 20-year period under review, some experts believed that fan design and performance had reached their zenith. In the last 10 years, however, there has been a substantial resurgence of design effort, coupled with manu-

facturing methods, better to streamline flow. Spinning and die-forming, together with newer welding techniques, have contributed to quieter performance and higher efficiencies, which users have loudly demanded.

The trend has been toward the backward curved blade centrifugal fan with its inherently higher speed characteristics. Where peak performance must be maintained with minimum horsepower, airfoil blade centrifugals are becoming important, despite increase in fan size and cost. Another advance has been to control fan performance with variable inlet vanes, which not only permit constant speed motors, but result in better efficiency at part load than by dampering at the fan outlet.

Great strides have been made in the use of axial flow fans. Their main applications are in low pressure applications where large volumes of air must be moved.

Air Distribution Equipment: There have been few improvements in the fabrication of conventional round and rectangular ducts made of metal, although there

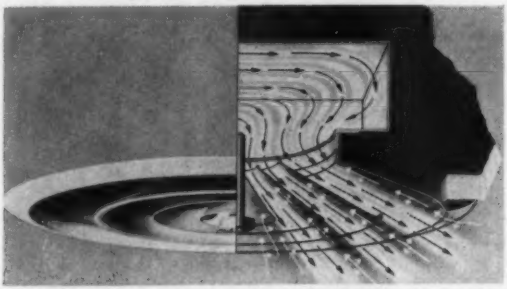


FIG. 19—CEILING DIFFUSERS DEMONSTRATED 20 YEARS AGO THAT AIR OUTLETS COULD BE PLEASING TO THE EYE AS WELL AS EFFECTIVE IN DISTRIBUTING AIR

has been increasing use of aluminum in place of galvanized steel. Special purpose ducts and conduits, with or without insulation and/or sound-proofing, have been introduced in substantial variety. A round spiral conduit with a watertight seam made by machine from continuous strip has proved especially suitable for high-velocity systems because of its low leakage rate, absence of distortion due to pressure, and savings in cost. Used with this conduit are special fittings, including ells, tees, and take-offs, suitable for multi-story application.

Composition ducts and conduits found considerable use during the war when steel and aluminum were unavailable. They have attained limited acceptance in residential and small commercial applications.

Knockdown ducts and conduits are a result of the recent expansion in the residential market, coupled with the *do-it-yourself* craze in home crafts. Round or square ducts complete with fittings are shipped in pieces for assembly by the man of the house. Some of these systems employ a multiplicity of small conduits rather than mains and branches of different sizes.

The mid-thirties saw the introduction of a great variety of room air distribution devices. Of substantial importance was the development of ceiling diffusers (Fig. 19) of pleasing appearance designed to distribute air uniformly and mix it thoroughly before introduction to the breathing zone. This was paralleled by similar work on

sidewall grilles. Where large volumes of air were to be handled draftlessly, the perforated ceiling distribution method filled the bill. Much more attention has been paid in recent years to control of air direction to suit the needs of the occupant. Well engineered regulation of volume and introduction of sound-absorbing material to cope with fan and regenerated noise have been important results of research since 1936. Design improvements have also been made to minimize dirt streaking of ceiling and grille surfaces. Finally, development of single and dual

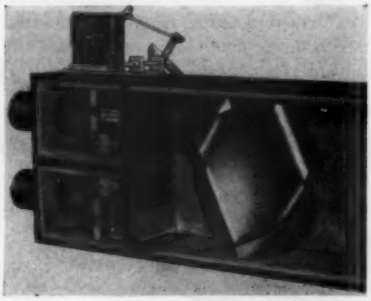


FIG. 20—THE ECONOMIC ADVANTAGES OF HIGH-VELOCITY DISTRIBUTING SYSTEMS NECESSITATED THE DESIGN OF AIR DIFFUSER CONTROL UNITS TO REGULATE AIR VELOCITY, VOLUME AND NOISE LEVEL. PICTURE OF DOUBLE DUCT CONTROL UNIT SHOWS (L TO R.) ENTERING DUCTS, MOTOR-OPERATED PROPORTIONING DAMPERS, MIXER, SOUND ABSORBING CHAMBER, AND OUTLET OPENING

air outlets with automatic dampering and noise control has been a recent important contribution.

HEAT EXCHANGE SURFACE

Heat exchange surface and equipment are important in any review of advances of air conditioning, since they represent 30 to 50 percent of the value of the end product.

By 1936 the industry had abandoned entirely the use of cast-iron blast heating surface and pipe coils, in favor of light weight finned coils, first marketed in the early twenties. The tubing was usually copper, and the spiral or plate fins were copper or aluminum.

Finned coils for cooling started to displace spray washers in the early thirties. Washers did not prove satisfactory in multi-story buildings, since the open spray system created hydraulic problems, required excessive pumping power, and was subject to contamination from impurities in the air. The closed chilled water

system employing finned coils offered measurable advantages in performance and cost. Another factor which promoted coils was the development of direct expansion systems employing the fluorinated refrigerants.

Early finned coils used soft solder for joining tubes and return bends. During the latter thirties high temperature bonding techniques were perfected using silver solder and brazing compounds, substantially increasing tube tightness and life.

An important development in heat exchanger design during the war was the sandwich coil, employing aluminum sheet stock without the use of tubing. This design was made practical through oven brazing techniques, whereby formed plates are

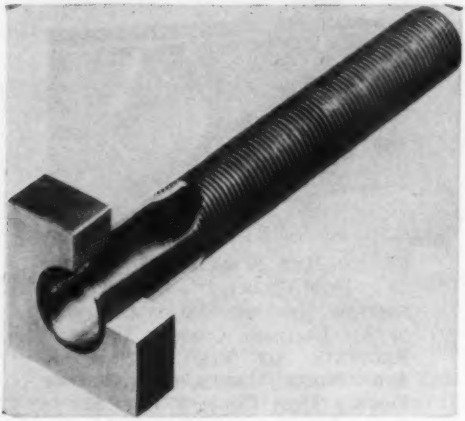


FIG. 21—PRESSURE-FORMED EXTENDED SURFACE TUBING FOR SHELL AND TUBE HEAT EXCHANGERS. THIS WAS ONE OF THE INDUSTRY'S MAJOR ENGINEERING ADVANCES OF THE PAST 20 YEARS

joined into a rigid box structure. To date these coils have been used primarily for aircraft heat exchangers.

During the past 10 years there has been a marked change in the choice of materials for fins and tubing. After the war, the price of aluminum relative to copper has steadily decreased on the basis of dollars per Btu transferred. Consequently, major research has been invested in developing all-aluminum heating and cooling surfaces.

A revolutionary development in heat exchanger surface was the introduction about 1937 of the pressure-formed-fin tube. This has had a tremendous impact on refrigerating practice because it cut in half the size of shell and tube coolers and condensers. The transfer rate of the new finned tube (developed by Carrier and Wolverine) is about 2½ times that for plain tubes. As shown in Fig. 21, the fins are about 1/16 in. high and number from 16 to 25 per in. They are pressure-formed from the tube itself, cut to length, the tube ends are belled out, and the

fin shaved off the belled end. The tube, which is rolled into the tube sheet in the same manner as bare tubing, may be removed in case of failure since the diameter over the fin is slightly less than the hole in the tube sheet. This type of tube ranks with the development of the centrifugal refrigerating machine and the fluorinated refrigerants as a major advance in the art.

A further trend in surface and coil design has been the optimization of geometry, which includes the number of fins per inch, fin thickness, tube spacing in both directions, the outside/inside surface ratio, and other physical optimums. Here the purpose has been to achieve the most economical usage of material, in order to transfer the greatest number of Btu per dollar invested. It is a characteristic of heat exchangers that the labor cost is small compared to the material cost. This has permitted considerable design and production ingenuity. The number of varieties of plate fin and spiral fin, not to mention interrupted fin, spike fin, and other types, is evidence of the effort and progress made toward reduction in material costs.

Refrigeration Equipment

Although refrigeration is not air conditioning, the rise of air conditioning to a multi-billion dollar industry would have been impossible without it. By similar token, refrigeration would still be in the dark ages were it not for air conditioning.

All the essential ingredients for design of both reciprocating and centrifugal compressors were known for many years prior to 1936. Progress during the past 20 years has not been in the nature of revolutionary change, but rather of improvement and refinement, with substantial cost reductions due to design economies and volume production.

Positive Displacement Compressors: Of the 2 types of positive displacement compressors—rotary and reciprocating—the rotary compressor can be dismissed rather briefly. Small rotaries, developed for refrigerators and using sulfur dioxide, have been redesigned and improved for use with low or high pressure fluorinated refrigerants. In the past 5 years there has been some activity toward design of larger fractional horsepower rotary compressors suitable for room air conditioners. Large methyl chloride rotaries developed 20 years ago are no longer built, but a more recent model is available as a booster compressor for low temperature refrigerant-12 staged systems.

The majority of positive displacement compressors are reciprocating. The most important evolution in these compressors has been increase in rotational speed. In 1936, compressor speeds had reached an average of between 400 and 600 rpm. Adaptation of automotive engine design principles made attainment of higher speeds contingent on improvement in pressure-operated suction and discharge valves for acceptable performance and life. The high-speed radial compressor introduced by Air-temp in the thirties spurred considerable research and development, but by 1945 there were several compressors available up to 100 hp operating at 1200 to 1800 rpm. Today much work is in process to perfect compressors to run at 3600 rpm, the maximum speed attainable by a 60-cycle two-pole motor.

Other design achievements included: (1) improvement of rotary shaft seals for open compressors to reduce service and maintenance costs; (2) pressure oiling systems, to insure adequate lubrication of moving parts; (3) improved system and piping design and installation techniques, to insure oil return to the crankcase, and

minimize the effect of contamination in the system; (4) improved manufacturing techniques with respect to cylinder surface finish, dimensional tolerances, alignment, and balance, contributing to longer life; and (5) capacity control methods.

All these improvements might be labelled *tail wags dog*. Prior to the '30's, compressor design progress followed commercial and industrial ice machine requirements. Since then, design technology has been stimulated intensely by the domestic refrigerator and air conditioning. Certainly more progress has been made in compressor design during the last 20 years than in the previous 50. The benefits of these improvements have been since returned to the commercial and industrial refrigeration field in terms of design improvements for their specialized machinery.

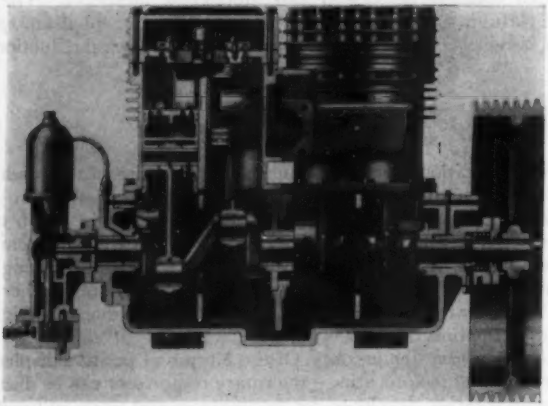


FIG. 22—BELT-DRIVEN RECIPROCATING REFRIGERANT-12 COMPRESSORS SUCH AS THESE WERE TYPICAL OF DESIGN PRACTICE IN 1936

Another inevitable trend, by no means new in principle, but relatively recent in acceptance, has been conversion of the industry to hermetically sealed units. Historically, this began first with domestic refrigerators, in General Electric's monitor top in 1927, so that by 1938 all new production of refrigerators by the industry was of the hermetic type. The next step included larger fractional horsepower compressors, and by 1954 room air conditioner compressors were entirely hermetic. Now there is a strong impetus to convert entirely to hermetic compressors in all sizes.

Motor winding and other components operating in a refrigerant atmosphere, possibly contaminated with water, air and oil, create difficult chemical and electrical problems. Much work has been required to overcome these problems, not the least of which was development of winding insulation more resistant to the contaminants. Sufficiently long life and freedom from maintenance have now been achieved to permit the hermetic compressor to supersede the open machine. The lower manufacturing cost of the hermetic permits another contribution by the industry in bringing the consumer a better and lower cost product.

Centrifugal Compressors: The principal change in centrifugal compressors in the last 20 years has been reduction in manufacturing cost through larger volume, improvement in performance, and hermetic design. It should be recalled that, although Carrier introduced the first open centrifugal refrigerating machine in 1921 and the first hermetic in 1930, it was not until the late thirties that 3 other companies entered the business. Additional companies have offered models in the last 2 years.

So far, the economic place of the centrifugal compressor has been in sizes well above 100 tons, although units are available down to 35 tons and up to about 3000

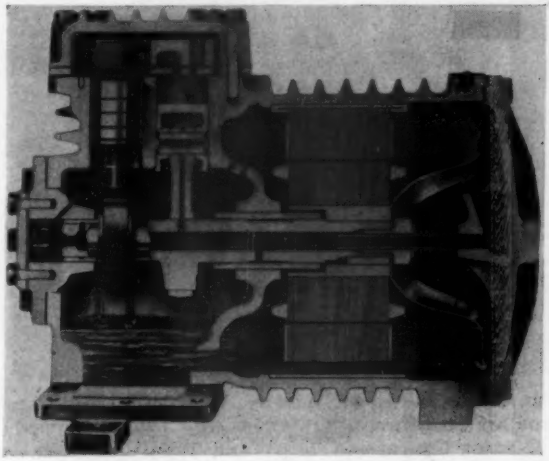


FIG. 23—MODERN HIGH-SPEED HERMETIC RECIPROCATING COMPRESSOR FOR ROOM AIR CONDITIONERS

tons. Compressors are typically one or two-stage for air conditioning. Above 300 or 400 tons capacity in a single unit, the centrifugal has completely superseded the reciprocating compressor, which up to 1937 was built in sizes for refrigerant-12 up to 700 tons.

In 1938 Trane introduced the first direct-driven hermetic unit, and during the past few years there has been a strong trend toward this type of machine, due to lower noise level, lighter weight, and ease of attaining fully automatic operation.

Water Chilling Units: For many years it has been an industry problem whether to supply components selected separately and assembled in the field to a tailor-made design, or to use water chilling units in which all components were designed and assembled at the factory. Field assembly has always been fraught with the hazards of inferior piping design and installation practices, which can be held under closer control with factory-assembled units. The past 5 years have seen a trend toward factory-fabricated and assembled reciprocating water chilling units. Complete units always have been furnished with centrifugals.

Heat-operated Refrigerating Machines: The aqua-ammonia absorption system was invented in 1815. Since that time there have been a great many attempts to im-

prove the absorption cycle, some of which have employed other refrigerant/absorbent combinations.

Most attempts to develop other types of heat-operated cycles have met with only limited success. Steam jet refrigeration has been often used for specialized applications, but has not enjoyed tremendous popularity for air conditioning because of the large steam consumption and size of cooling tower required.

Another attack has been made by using internal combustion engines burning natural gas to drive compressors in a conventional compression cycle. While a number of installations were made during the thirties, high maintenance costs,

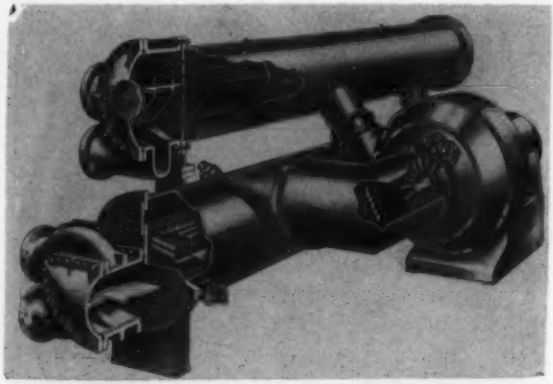


FIG. 24—HERMETIC CENTRIFUGAL WATER CHILLING UNIT FOR AIR CONDITIONING (1956). SHOWING DIRECT-DRIVEN, 2-STAGE COMPRESSOR, SHELL AND TUBE CONDENSER (ABOVE), AND EVAPORATOR (BELOW)

noise levels, and other objectionable features resulted in decreasing use. During the last 3 years there has been a renewed effort to solve the need of the gas utilities for a summer gas air conditioning load. Inasmuch as gas engines have good operating economy, considerable development expense is being invested to improve reliability, automatic starting, and control, lengthen operating life between overhauls, and reduce noise to an acceptable level.

In 1937, Williams Air-O-Matic developed a water cooling unit for air conditioning using halogenated compounds in capacities of 15 to 35 tons. This machine was marketed for a few years, but was discontinued about 1952. In 1940 Servel introduced an absorption air conditioning unit employing water as the refrigerant, and an aqueous solution of lithium bromide as the absorbent. The operating efficiency of this combination is substantially better than the aqua-ammonia cycle. These hermetic water-cooled machines were introduced as all-year heating and cooling units for residences and small commercial installations, in capacities of 3 to 5 tons. Mechanically, great ingenuity was used in overcoming the problems of a high vacuum machine using water as the refrigerant. In recent years strides have been made towards simplification and cost reduction through direct firing and use of the same coil for both heating and cooling.

In 1945, Carrier introduced the first commercial lithium bromide absorption water chillers for air conditioning, in capacities from 100 to 700 tons and operating on low pressure steam. Completely automatic, these units adjust to any load from 0 to 100 percent, and their low noise level and freedom from vibration permit installation wherever space and steam are available. Recently, Servel introduced a 25-ton water chilling unit.

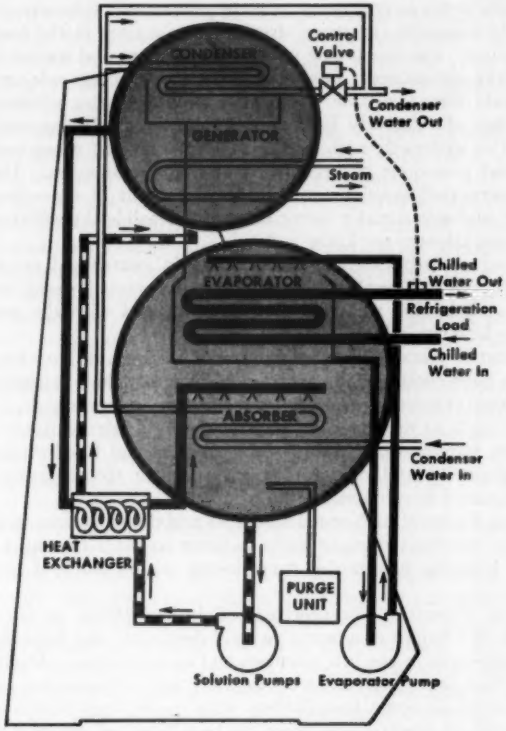


FIG. 25—CYCLE DIAGRAM OF LITHIUM BROMIDE ABSORPTION WATER CHILLING UNIT FOR AIR CONDITIONING

The absorption machine offers substantial operating cost advantages in many areas, and permits utilization of otherwise idle boiler capacity during the cooling season. The author predicts that the development of the lithium bromide/water absorption machine will be viewed in retrospect as one of the really important advances in the technology.

Refrigerants: The outstanding development in the field of refrigerants during the preceding 20-year period was the invention of the "Freon" family of fluorinated

hydrocarbon materials by Midgely and Hene in 1931, and their subsequent development by Kinetic Chemicals. While refrigerant-11 and refrigerant-12 started to gain commercial importance in centrifugal and reciprocating compressors respectively prior to 1936, their real impact on the technology of the industry has occurred during the past 20 years.

For reciprocating machines, the fluorinated refrigerants spelled the death knell to methyl chloride, carbon dioxide and sulfur dioxide, in all categories of equipment from refrigerators to the largest units. Of the refrigerants formerly of commercial importance, only ammonia continues strong today, mainly in the food processing and storage industry, due to existing investment in plant and inertia in taking advantage of modern techniques. For the first time, completely safe and acceptably stable refrigerants were available which gave good operating efficiency and permitted the design of relatively light weight machines. Design techniques had started along this approach with methyl chloride several years earlier, but its flammability and poisonous nature greatly limited application. The advent of refrigerant-12 permitted development and improvement of these compressors to proceed swiftly, and was a major factor in making feasible the self-contained units and room air conditioners we know today.

The important contribution of refrigerant-11 in centrifugal compressors was toward cost reduction. Together with other improvements it permitted to be done in 2 stages of compression what required 5 or 6 stages with the predecessor refrigerant, methylene chloride.

During the past 20 years many other fluorinated refrigerants have been developed, and a few have been commercially successful. These include refrigerants-113 and -114 for centrifugal compressors and 22 for reciprocating compressors. Low boiling point materials such as refrigerants-13 and -14 were made available for low temperature process refrigeration. In 1946, Pennington and Reed invented an azeotrope of refrigerant-12 and difluoroethane for use in reciprocating compressors (presently designated as refrigerant-500).

By permitting design of high-speed machines and direct expansion systems these new refrigerants contributed importantly to lower costs for the entire refrigeration cycle, thereby bringing summer air conditioning within reach of more and more people.

Dehumidifying Processes: The processes of dehumidifying air by contact with moisture absorptive brines or adsorptive solid dessicants had been developed into commercial equipment in the late twenties and early thirties. During the period under review, they proved not to be competitive with refrigeration for comfort air conditioning, although some installations were made; but they found a special niche in industrial air conditioning where low humidities are required for product or process. Major strides were made in improving equipment performance and durability.

WATER CONSERVATION EQUIPMENT

During the early days of air conditioning the pressure to conserve water was not particularly great. While large installations had for some years used cooling towers, small air conditioning plants used city water directly and disposed of it to the sewer.

With the development of the fluorinated refrigerants in the early 1930's, volume marketing of small air conditioning units began and water conservation became imperative. In many areas local ordinances were enacted requiring conservation equipment even for small users, and in many other localities the cost of water be-

came enough so that it became profitable to save it despite the heavier capital investment. In some cities sewers were overloaded, resulting once more in regulations forcing water saving devices.

None of the 3 major types of water conservation equipment is new in principle. The shifting trends in their use reflect mainly design improvements and modifications, and heavier economic pressures.

In cooling towers, there has been a trend toward forced and induced towers and away from gravity types with their uncertain performance and large size. Development of cross-flow fills have reduced tower height significantly, and progress has been made toward lower noise levels and reduced water carryover losses. In the realm of small towers much effort has been placed upon manufacturing standardization, cost reduction, and increase in durability.

The evaporative condensing principle, whereby water is sprayed upon the condensing surface and heat is dissipated by evaporation of part of this water into air, had been used for many years with gravity ammonia pipe coils. In the early 1930's forced air evaporative condensers were employed in steam ejector cooling units in railway passenger cars. The principle was adapted to refrigerant condensers in the mid-'30's, first by modifications of fan coil units, and later in built-up sprayed coil assemblies. Compared with a cooling tower, the evaporative condenser usually permits significant space and cost savings. Its use however has been confined to smaller tonnages and high pressure refrigerants, due to the problems and cost of running large inter-connecting refrigerant lines between compressor and condenser.

Contamination of recirculated water has proved to be an inherent problem with both cooling towers and evaporative condensers. Metallic parts become corroded, scale is deposited from hard water, and frequently algae accumulate objectionably. During the past 20 years a great deal has been learned about water treatment. At the same time, water problems have provided a stimulus for the development of the dry surface condenser.

Dry condensing is not new either, although its popularity is relatively recent. Substantially larger than either the water cooled or evaporative condenser, it frequently offers improved system costs wherever water must be saved, and definitely lower maintenance costs and less operating nuisance. The normal spring and fall service calls to start up and shut down for the season are eliminated. The air condenser has been used in room air conditioners for 20 years. Within the past 5 years, its growth into the small residential and commercial air conditioning market has been astounding, from practically zero in 1951 to more than 50 percent of the market today. For 24-hour operation, as in residences, operating costs are competitive with cooling towers and evaporative condensers, due to the drop in outside temperature at night.

Automatic Controls

Improvements in automatic controls during the past 20 years have occurred in many directions. They include greater sensitivity of thermostats and controllers, true modulation and positive positioning of control valves and dampers, and better sizing and port design of control valves. The size of valve and damper actuators has been reduced and their power increased. In response to consumer demand, the appearance of thermostats and other in-space controllers has been greatly improved.

A wider variety of control devices is now available. Better understanding of their application has resulted in pin-pointing their selection to achieve closer control of temperature and humidity. There has been an increasing trend toward individual room control. Other improvements include automatic use of outside air for

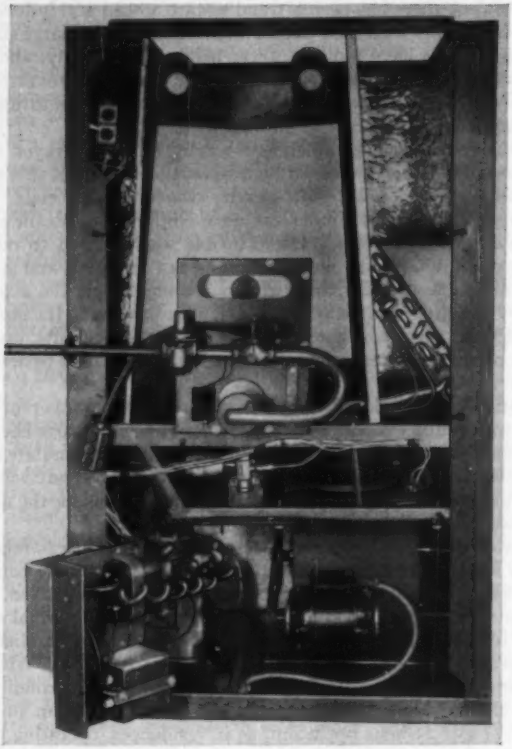


FIG. 26—INTERIOR VIEW OF 2-TON CENTRAL PLANT UNIT FOR THE ALL-YEAR AIR CONDITIONING OF RESIDENCES, SHOWING FUNCTIONAL ELEMENTS

cooling economy, and solar and outdoor compensation for building exposure to sun and wind.

Sensitivity, accuracy, simplicity and reliability have been improved by the introduction during the past 8 years of electronic sensing and control. The broader use of remote indicators and control devices has encouraged the use of data control centers where a lone operator may check temperatures, pressures and relative humidities in all parts of a building, start and stop fans, compressors and pumps,

and readjust dial settings as required. Better manufacturing, inspection, and testing techniques have improved the cost and reliability, and have reduced maintenance.

AIR-CONDITIONING SYSTEMS

No examination of air-conditioning progress would be complete without a review of the systems which have resulted from the application of our basic knowledge and array of modern products.

Residential Systems: During the past 5 years, the rapid acceptance of residential air conditioning has opened up new horizons in the design of central plant self-contained air conditioners. Residence application is quite different from commercial, for the equipment is concealed and may be located in the basement, garage, utility closet, attic, or outdoors. With a volume market in prospect, manufacturers have designed packages specifically applicable to each of these locations. The trend toward air cooling with its need for location near a source of outside air has given rise to the 2-package or 3-package air-conditioning plant. Space does not permit listing the different combinations of packages now available. All fall into the class of either all-year or summer conversion equipment.

The all-year package is designed primarily for new construction or where the existing heating system is to be replaced. It is usually located in the basement or in the utility closet, and may be served by a cooling tower, or may have a separate air-cooled condenser package.

The add-on conversion cooling unit is designed specifically for addition to an existing heating system, but it is also frequently used in new construction. Here the trend is toward remote location, frequently outdoors, with only the cooling coil (and fan if required) located near the furnace.

Most of these types employ direct expansion cooling. However, there are also available self-contained water chillers which may be located outdoors with suitable freezing protection. This equipment may be advantageous in conditioning homes supplied with wet heat, by substituting suitable fan coil units for existing radiation. Smaller houses with wet heat are often cooled successfully by direct expansion units in the attic.

The importance of the room air conditioner in the residential market has already been noted. But it is proving to be a calling card, a preview of the benefits which air conditioning can bring into an entire home. More often that not, the best and most economical solution proves to be a central system.

The heat pump, too, has been described in the section devoted to equipment. It may yet have a unique place in the home, but it has to justify its glamour through solid performance and demonstrated superiority in the face of higher investment and operating costs.

Finally, the gas utility industry has at long last recognized the challenge of the electric heat pump and the all-electric home. It is sponsoring strong research and development programs designed to make equipment burning gas as much an economic factor in residential cooling as it has been in heating. The all-year absorption unit has been available to the home market for many years. Now in the past 2 years have appeared compression units powered by natural gas-driven internal combustion engines, and field samples of new types of fuel-operated cooling cycles. Truly the art is moving faster with every passing day.

A relatively recent market lies in mobile homes (house trailers), where the obvious first approach was to apply standard room air conditioners. Space and ar-

rangement problems proved this course a makeshift, and special-purpose units are now available.

Commercial Systems: Perhaps the most important single system developed during the past 2 decades is the high-velocity system, engineered to overcome the problems of multi-room air conditioning. It was created from sheer economic necessity. The costs of constructing multi-story, multi-room buildings have risen to a point where space conservation for service functions is a major consideration. Moreover, maintenance costs have risen to a point where owners are hypersensitive about overhead.

The conventional air-conditioning solution to the problems posed by this type of building involved a central air-conditioning apparatus which propelled large volumes of conditioned air into each room at low velocities through big supply ducts. More ducts, equally large, were required for air return. This straight-forward engineering approach produced acceptable results, but sacrificed valuable floor space, headroom, and shaft area.

Unlike theaters or department stores, multi-room buildings, such as office buildings, apartments, hospitals and hotels, have a high percentage of outside rooms, with shifting solar heat gain as much as 60 percent of the total sensible load, and internal load varying from room to room with lighting and occupancy. The conditions maintained by a single zone temperature controller, perfectly acceptable in a large space such as a theater, were often unacceptable in multi-room buildings where some rooms required heating and others cooling. Also, individual control by the occupant was fast becoming a basic requirement. The solution of this problem by Willis H. Carrier in 1937 embodied an entirely new approach. It did not attempt to circulate all of the air required for cooling, but only that minimum necessary for ventilation. Rather than processing 100 percent of the air circulated in the building, his conduit induction system preconditioned only that 25 percent represented by the ventilation air, and utilized it as the motive power to induce secondary air over coils in the induction units located in each individual room.

Because less than one-fourth of the air was circulated, small conduits were employed to distribute it at velocities of 3000 to 4000 fpm from the central primary air conditioner to the room units. In the latter the primary air passed at high velocity through nozzles inducing room air over the coil supplied with either heated or cooled water, and thence into the room.

This system represented an important milestone, because it fulfilled an urgent economic need—the need to save valuable space. Moreover, this saving was achieved without sacrifice of any important performance objective. Since ventilation, room humidity and circulation were stabilized because of the total absence of dampering devices in the air circuit, individual room temperature control became a reality.

The conduit induction system represented the first modern primary air system for multi-room buildings. Other popular variations employed fan coil units in each room to provide the sensible heating or cooling. These systems were functionally identical with the induction system except that a fan replaced the induction nozzles. An inferior but widely used arrangement employed fan coil units without a central primary system.

An innovation of the last 5 years was the dual-duct system, which had been borrowed from schoolhouse practice of 50 years ago, and streamlined to meet the needs of modern multi-room buildings. It required no water piping, but produced its cooling or heating by mixing air from 2 ducts, supplied at different temperature

levels. While the cost of water piping was saved, the conduits, even with air at high velocity, were much larger than with the primary air induction system, and therefore fewer floors could be served by each air-conditioning unit. Although it represented a considerable space saving over the conventional low pressure system, the dual-duct system by definition implied variation in air volume at each distributing element, resulting in control complications. Nevertheless, progress has been made in solving the inherent problems of instability, pressure regulation, noise, and air distribution.

In the effort to avoid large ducts, another technique of importance developed in the last 20 years was the distributed unit system, which employed relatively small

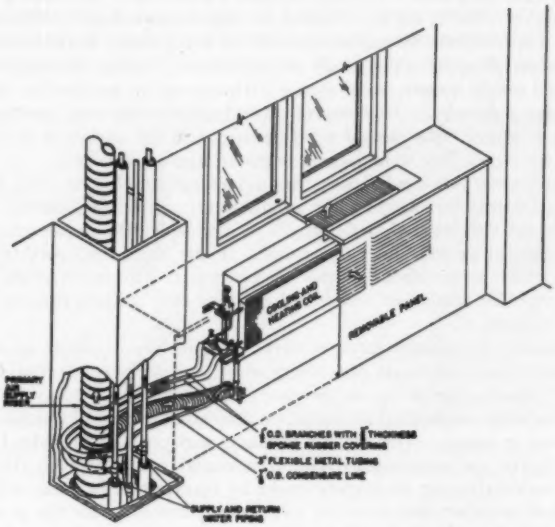

COOLING AND HEATING COIL
REMOVABLE PANEL
O.D. BRANCHES WITH THICKNESS SPONGE RUBBER COVERING
FLEXIBLE METAL TUBING
O.D. CONDENSATE LINE
SUPPLY AND RETURN WATER PIPING

FIG. 27—CONDUIT INDUCTION UNIT, SHOWING (LEFT) PRIMARY AND SECONDARY AIR FLOW, AND (RIGHT) CUT-AWAY VIEW FROM FRONT

air-conditioning packages to serve the various zones of a multi-room building. These were normally supplied with chilled water from a central refrigerating plant located in the basement. A more recent trend has been the use of self-contained units within each zone, thereby eliminating the network of chilled water pipes from the central unit. Only cooling tower water need be piped to the units.

Beyond the multi-room systems just described, the advances in system techniques during the past 20 years in conditioning commercial buildings were mainly refinements in design and installation. Prompt advantage was taken of new designs in equipment.

There were 3 important improvements in the air conditioning of theaters and auditoriums. One was better air distribution, particularly in the larger houses,

through better engineering and diffuser selection. Another was improved silencing of the air conditioning through better isolation of machinery and sound attenuation in the ductwork. Finally, there was vastly improved operation of equipment by theater management, with less indoor-outdoor temperature contrast. The day of the paper icicles on the marquee had passed, and this indeed was evidence that air conditioning was coming of age.

Small stores and many large ones have taken advantage of vastly improved equipment to make possible space saving and equipment decentralization. In the larger stores, the trend has been away from space-consuming central systems with large shaft and duct areas, toward decentralized units or assemblies hung from the ceiling, located on mezzanines, or placed in the conditioned area but occupying minimum floor area. During the past 5 years there has been a trend toward much higher velocity air distribution for department stores, with consequent space saving, improvement in appearance and, in many cases, lower costs.

Other commercial applications, such as restaurants, banks, monumental buildings, and hotel public spaces, have shown little change in application techniques during the past 2 decades. In many of these buildings the chief problem is the structure itself, rather than the air conditioning, and the approach is necessarily dictated by cost, feasibility of making alterations, and appearance.

Industrial Systems: Air-conditioning techniques for the 200 or more industries that were responsible for creating the original air-conditioning market 30 to 40 years ago had been developed and brought to a considerable degree of perfection prior to the 20-year period under discussion. There have been very few *new* industries into which air conditioning has been introduced in recent years, although certainly many older industries which had received only a slight impact have now become heavy users.

Most technological changes have revolved around improvement in equipment and availability of new methods, plus the great acceptance of comfort air conditioning *per se*. This acceptance has in the past decade motivated many industries to install systems with mechanical refrigeration instead of evaporative cooling. Textiles are a case in point. The specific products and processes required humidity fixed within limits, and temperature was often secondary. However, the only way conditions comfortable for employees could be maintained in these mills during normal summer weather, simultaneous with proper humidity for the product, was through the use of mechanical refrigeration.

While many industrial applications where close humidity control is necessary continued to use large spray humidifiers and dehumidifiers, an increasing number have been buying unitized systems employing smaller building blocks of capacity. As products were perfected with respect to endurance and maintenance, there was less objection to numerous sets of equipment scattered around a plant compared with fewer and larger central systems. This has turned out to be largely an economic problem, involving many factors. The answer is often different for different types of buildings and processes. The number of installations has been increasing using factory or field-assembled complete air-conditioning and refrigeration cycles, ranging from self-contained equipment to units of 75 to 100 horsepower. On the other hand, the operating advantages of plants with large centralized units of water chilling capacity have also been evident.

Marine and Naval Systems: In 1936 the only air conditioning on ocean liners consisted of public rooms such as dining saloons. Since the end of the second World War, nearly every passenger ship commissioned has included air conditioning in all

staterooms as well as public rooms, and usually in crew's quarters. In recent years this has also included cargo vessels.

Because of space limitations, marine installations have traditionally employed velocities of 3000 to 4000 fpm. Recent applications have edged up to 5000 fpm with a few reported abroad as high as 7000 fpm.

Frozen food storage requirements have affected the design of cargo conditioning rooms. In 1936, cargo vessels typically used 15 F, but today zero and −10 are not uncommon. Also there is a marked trend to direct expansion systems, although brine is still used. In 1936 precooling of cargo in a dockside warehouse was conventional practice. Since the war there has been a shift to the *hot* cargo method, which requires larger equipment but avoids precooling altogether. Finally, there has been a trend toward temperature control of each individual space. Manual control of brine flow has given way to automatic temperature control.

The first application of air conditioning to naval ships in this country was aboard submarines in 1933, both for comfort and to assure proper operation of the elec-

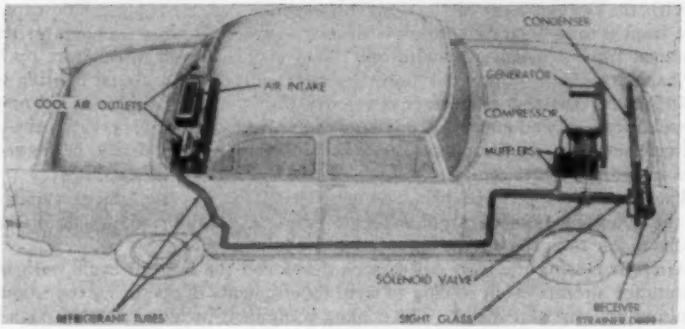


FIG. 28—FACTORY INSTALLED AUTOMOBILE AIR CONDITIONING SYSTEM IN A 1956 MODEL, SHOWING COMPRESSOR BELT-DRIVEN FROM MAIN ENGINE, COOLING COIL BACK OF REAR SEAT

tronic equipment. But it was not until the advent of World War II, when surface vessels became compartmented and the quantities of fire-control and heat-producing equipment vastly increased, that air conditioning came into general use within those spaces.

The South Pacific engagements proved that it was necessary to install air conditioning in crew quarters, since without it as many as 10 percent of the crew were immobilized by the effects of high temperature and humidity. The other spaces usually conditioned included aviators' ready rooms, combat information centers, fire-control rooms, steering engine rooms and sick bay areas.

Automotive Vehicles: The 2 major problems in automotive air conditioning have been space and drive. Inter-city buses began to use air conditioning in 1938 with a system capacity of about 4 tons. High speed aluminum reciprocating compressors have replaced belt-driven cast-iron units, and recently aluminum coils have started to replace copper. Most frequently, compressor and condenser fan have been driven by a separate gasoline engine rather than from the bus engine, permitting

the capacity of the system to be independent of vehicle speed, but adding to the cost.

Development of automobile air conditioning has been retarded by the drive problem. So far all systems have employed some type of power take-off from the engine, and have run up against the well-known problem of obtaining adequate capacity at all vehicle speeds from idling to 100 mph. One recent design has integrated the cooling equipment under the hood with the car heater. Many early and current designs placed the cooling equipment in a separate compartment in the trunk or behind the back seat.

Railway Passenger Cars: During the thirties, several thousand reciprocating compression, steam ejector, and ice-melting systems were installed. Both of the latter have now been superseded by the lighter and less costly reciprocating system, powered by storage batteries charged from an axle-driven generator. For a number of years, many Pullman cars employed compressors driven from the car axle through a special eddy-current clutch, but this system was abandoned because of excessive power required at high speeds.

With the development of the dome observation car, air-conditioning capacities rose from 8 to as high as 16 tons. Since the power required was greater than is practical from axle-driven generators, diesel engine driven alternators up to 40 kw capacity supply power for both the air conditioning and the lighting loads. With some of the new low center of gravity trains of the Talgo type, all power is supplied from a head-end diesel engine-driven alternator. This permits the use of self-contained air-conditioning units on each car with electric-driven hermetic compressor.

There have been no radical changes in air distribution systems on cars since 1936. In the older cars, the supply ducts were mounted on the roof outside of the car. In the newer cars, the duct is mounted above a false ceiling.

Aircraft: During the war, AiResearch developed the first air cycle refrigeration system for aircraft cabin cooling to rigid requirements of operating conditions and minimum weight and space. The cooling equipment included an ultra-high-speed centrifugal air compressor, an air expansion turbine, an aluminum sandwich-type air-to-air heat exchanger, and special controls integrated with cabin heating and pressurization. This type of equipment has since achieved wide use in both military and commercial aircraft.

In the post-war period, electric-driven vapor cycle refrigeration systems first have been installed on a number of commercial airliners. Unlike air cycle, these systems could operate while the plane was on the ground by plugging in a power line. In order to meet space and weight limitations, substantial design modifications were required in conventional reciprocating compressors, evaporators and condensers. The recent advent of turbo-prop and turbo-jet transports has permitted use of turbine engine bleed air as the source of pneumatic power to drive vapor-cycle centrifugal compressors. Weight of the Carrier 15-ton system for the DC-8 is only 150 lb, of which the high speed turbo-compressor unit accounts for less than 10 lb.

FUTURE HORIZONS

In future years, air conditioning will develop not alone through technical progress in this industry, but even more importantly through the scientific and engineering

advances of our whole economy. Let's stop and gaze into the crystal ball. It may be somewhat clouded, but the picture is truly challenging.

The air-conditioning industry itself will evolve along well-known economic patterns. As it becomes more firmly established on the plateau of maturity, manufacturers will become larger and stronger. Competition will accelerate the trend toward integration to reduce manufacturing costs, and the consumer will be the beneficiary.

Manufacturers will achieve greater integration of their product lines for maximum interchangeability. They will make an intensive effort to reduce equipment

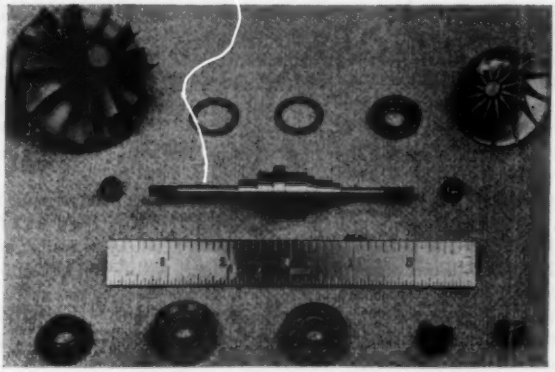


FIG. 29—FUNCTIONAL COMPONENTS OF 15-TON TURBO-COMPRESSOR FOR AIRCRAFT CABIN COOLING. COMPRESSED AIR-DRIVEN TURBINE WHEEL IS AT LEFT, REFRIGERANT-12 COMPRESSOR WHEEL AT RIGHT. NOTE 6-IN. SCALE

size, and strong efforts toward miniaturization will surely be seen in the near future. Military requirements for aircraft and missile cooling will inspire much of this. The early steps imply cooling and heating coils designed to transfer much more heat in much less space. Later, tiny centrifugal compressors can be expected whirring away at fantastically high rotative speeds, driven by motors which employ electrical principles still but a gleam in the inventors' eyes. Radically new air-moving devices will feature an absence of noise and moving parts. And ultimately the product will be inexpensive enough through mass production so it will be cheaper to replace it than repair it when it wears out.

In the future, development of the Man Cooler can be visualized carried in the hip pocket like a walkie-talkie, delivering personal cooling at the flick of a switch. As the air-conditioned man of tomorrow walks around in the broiling sun, solar batteries will generate the power to cool the thermoelectric clothing he is wearing.

While the Man Cooler concept may represent an ultimate and eventual extreme, sensational advances can be expected in environmental control much sooner. The improvements today in air cleaning techniques and products are ahead of consumer realization and acceptance. Practical air sterilization methods are at

hand, awaiting further development. Smaller and cheaper odor control devices will permit reduction in outside air to the tiny amount needed for oxygen replenishment and CO_2 removal. Production of desirable odors, a form of psychological environment control, may become commonplace.

This leads to all kinds of intriguing possibilities involving the future therapeutic uses of air conditioning. For example, oxygen added to air appears to create a sense of well-being, and properly controlled, may improve man's health. Its medical importance to persons suffering respiratory and functional disorders is common knowledge. Why should oxygen be confined to the oxygen tent and the iron lung?

Environmental control of ions is claimed to produce similar sensations of well-being. Here is a frontier for immediate research, because the development of equipment for producing and controlling ionization has outrun knowledge of the true physiological importance of ions themselves.

Why stop at oxygen and ions? Why shouldn't medical science come up with answers as to what materials can be added to the air which will stop hay fever, work up an appetite, cure a hangover, dry up that cold, or put one in the mood to fill out an income tax return?

Let's look more closely at the 2 great markets of the future—the factory and the home.

Before the close of the next 20 years, most factories, old and new, will be air conditioned. If underground plants come into general use, they will have to be cooled, and if civil defense is the objective, they will need protection against poison gas and radioactive fall-out. Specialized equipment will be needed for spot cooling in hot areas and occupations, and factory employees may often carry their own. More attention will be paid to cancellation by insulation or ventilation of large sources of heat right at the source, whether due to sunshine from the outside, or lights or machinery on the inside.

Air conditioning of homes will continue its strikingly vigorous evolution. Many of the changes will be stimulated by prefabrication, be it of the entire house or of particular building blocks. Integration into a unit services package now seems a possible direction, to include refrigeration, freezer, stove, and hot water heater. And a panel in the outside wall will form the condenser to get rid of the heat from the refrigerating machinery.

Increased developments of all forms of transportation air conditioning can be anticipated. Hard-headed business men expect one-third of all new automobiles to be cooled by 1961. Subway cars are being actively estimated right now, despite the technical and economic problems of cooling 200 people crushed together like sardines. In aircraft cooling where space and weight are controlling, the greatest design changes will be higher rotative speeds and more efficient use of lightweight materials. Speeds up to 10,000 rpm appear attainable with reciprocating equipment, much higher with centrifugals and rotaries. Classified work is being pushed on cooling supersonic military aircraft and guided missiles where flight speeds may produce missile skin temperatures up to 800 F. Some authorities believe that aircraft cooling of the future may require up to one-third of the energy produced by the main powerplant. And today's bombers carry power plants developing 100,000 hp. This line of thought leads to speculation about space satellites revolving around the earth, and space vehicles to other planets, and these will need some fancy air conditioning indeed.

Turning to the great outdoors, air conditioning will achieve spectacular results both in the city and on the farm. Inflated plastic canopies may be used to suspend

conditioned air over baseball and football stadiums, a forerunner of the air-conditioned street and the air-conditioned city. Air-conditioned pigs and chickens will produce more pork and more eggs per pound of feed. Especially in the tropical countries of the world, air conditioning will increase human productivity. And in the more distant future, total climate control might cause the migration of entire populations into today's undesirable regions of the globe.

Much of the future will depend upon the success of scientists and engineers in developing new sources of energy production and new sinks for its disposal. The imminence of new prime movers—gas turbines, free piston engines, hot air engines—should be recognized and their impact on economy and industry assessed. Nuclear and solar energy represent untapped sources today, tremendous in quantity compared to fossil fuels. Their transformation into usable heat and power hold great promise for further extension of both absorption and compression refrigeration cycles for cooling. Their transformation through as yet unknown photo-chemical, electro-chemical, thermo-electric or nuclear-electric processes would be revolutionary to existing technology.

The near future promises large central heat and power generation from nuclear energy as a certainty. Is it too much of a stretch of the imagination to visualize solar absorbers installed in the roofs of buildings to furnish the energy for heat pumps which supply the heating and cooling? Or to utilize the heat from the molten core of the earth, or the cooling from the icy depths of the ocean?

Well, of course, many of the foregoing projections are sheer flights of fancy, the stuff which dreams are made of. But some of these dreams are going to come true. The majority of them, while a long way off, are being dreamed by our most creative and knowledgeable scientists and engineers. And few of them are more unlikely than the actual facts of the progress of air conditioning during the last 20 years.

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1594

No. 1594

REVISED WINTER OUTSIDE DESIGN TEMPERATURES

By H. C. S. THOM*, WASHINGTON, D. C.

A WINTER outside design temperature may be defined as an outside temperature which, when used in the design of a heating system, will assure with a given contingency that the system will adequately meet a set of prescribed design requirements. It is clear that the choice of outside temperature for design purposes depends on the assignment of the prescribed design requirements and on the contingency involved. The assignment of the prescribed design requirements is an engineering problem while the determination of the contingency is a climatological problem. If there were no contingency associated with the climatological variable, the design problem would involve only the contingency on the engineering factors and this could usually be made small without increasing the engineering factors beyond reasonable economic limits. Thus, the main contingency in the design of heating systems arises from the variability of the weather. This must be handled by climatological analysis.

There have been a number of published studies of the outside design temperature problem. These will not be discussed in detail here since the basic hypotheses used in this paper are believed to be somewhat different from those previously considered. This does not mean that the design values proposed here are unrelated to those previously discussed, but only that the proposed design values are believed to be more rationally related to design experience and, in fact, are governed to a considerable extent by that experience.

THE DESIGN VARIABLE

For purposes of generalizing the discussion, use will be made of the concept of a design variable. (See Reference 1 for discussion of another application). This is defined as a variable quantity which is rationally related to the design problem and can vary over a scale of contingency and a range of the prescribed design requirements. A particular value of the design variable under consideration would

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be a chosen design temperature. Clearly this temperature depends on a specific value of the contingency and on the prescribed design requirements. The problem of developing a design variable consists then in finding a variable which is rationally related to the elements of the design problem, and in defining functions or relationships on the design variable which relate it to the contingency and the prescribed design requirements. Hence, given a set of design requirements and a contingency on adequacy, these may be related to the design value.

The foregoing discussion may be more clearly expressed by using the language of function symbolism. The prescribed design requirements are summarized in the inside design temperature. This may be expressed as

$$t_{iD} = j(a, b, c, \ldots)$$

where

$a, b, c, \ldots$ = any set of design requirements.
t_{iD} = an inside design temperature.
j = a function relating t_{iD} and $a, b, c, \ldots$

Associated with t_{iD} is a contingency or probability, q, which is the probability with which a failure of the system to meet the load accompanying t_{iD} can be tolerated. We shall assume that t_{iD} and q are given since their determination is beyond the scope of this paper.

Let t_i be any inside temperature which is produced when the heating system is operating against an outside temperature t_o. There are conventional engineering methods which make possible the relating of t_i to t_o by a function, say g. This may be expressed by the functional relationship

$$t_i = g(t_o) \quad \ldots \quad (1)$$

For a specific inside design temperature t_{iD} there will be an outside design temperature t_{oD} given by Equation 1 in the relation

$$t_{iD} = g(t_{oD}) \quad \ldots \quad (2)$$

This may be determined explicitly by the inverse of Equation 2 given by

$$t_{oD} = g^{-1}(t_{iD}) \quad \ldots \quad (3)$$

Now by climatological analysis a probability distribution, F, may be defined on t_{oD} such that p is the probability of an outside temperature t_o being less than t_{oD}. This is given by

$$p = F(t_{oD}) \quad \ldots \quad (4)$$

Substituting the value of t_{oD} from Equation 3 into Equation 4 gives

$$p = F[g^{-1}(t_{iD})] \quad \ldots \quad (5)$$

Taking the inverse it is found that

$$g^{-1}(t_{iD}) = F^{-1}(p) \quad \ldots \quad (6)$$

and

$$t_{iD} = g[F^{-1}(p)] \quad \ldots \quad (7)$$

Thus it is seen that each t_{iD} is associated with a p on t_{oD} and if p is set equal to q (as defined earlier in this paper), the probability of an outside temperature being

less than t_{0D} is equal to the probability of the inside temperature being less than t_{iD}. Hence Equation 6 determines an outside design temperature t_{0D} by means of a probability on t_{iD}.

While Equation 6 gives a simple symbolic solution to the design temperature problem, the real situation is somewhat more complicated. Actually the function $g(t_0)$ is a simplification of the real situation. Although g is ordinarily assumed to be a function of a t_0 which is constant long enough to allow t_i to attain an equilibrium value, actually t_0 is always varying with time. Since t_i lags behind t_0 because of heat storage in the structure and t_0 is often at its minimum only a short time, t_i may not reach the minimum defined by $g(t_0)$. Hence, in general, $t_i \geq t_{iD}$ when $t_0 = t_{0D}$. This has the effect of making $q < p$; or if p is set equal to q, the actual inside design temperature will be somewhat greater than given by the function g when heat storage of the structure cannot be taken into account.

Because t_0 has a diurnal cycle as well as an annual cycle, the exact definition of t_{0D} is a complex problem. If, however, the function g accounted for all factors in the relation between t_i and t_0, the design variable could be defined much more exactly. Unfortunately this would be a very complicated function which is not available in exact form at the present state of engineering science. Hence, in order to arrive at a generally useful set of design data, certain simplifying assumptions must be made. The complexity of g may then be allowed for by giving a range of design values, each with its associated probability, from which the engineer may for most practical purposes account for a known form of g by simply choosing a probability which fits specific conditions. For example, if the heat storage of a structure is high, a higher t_{0D} and hence a higher probability may be chosen. If it is low, a lower t_{0D} and a lower probability would be chosen.

The first simplifying assumption (assumption I) made is that the greatest stress on the heating system will occur when the average temperature, for a day or a linear function of it, assumes its lowest value. This implies that if the heating system meets the load satisfactorily for the *day* with the lowest average temperature, or a linear function of it, it will also meet it for every other day during the year in which that *day* occurs. For example, suppose that in 3 particular years the coldest days as measured by the average temperature were January 15, February 10, and January 25, and a heating system met the load on all of these days, then it must have also met the load on all other days during the 3 years.

Observe that the assumption is not very restrictive, for as stated, the designer may make adjustments by varying probabilities. Note also that this clearly segregates the operating problem.

The second simplifying assumption (assumption II) is that the design temperatures in use for a good many years[1] are correct on the average over the country for a given contingency on the satisfactory design of heating systems, or if not correct, all are in error by a fixed probability difference. A consideration of these 2 assumptions together leads to a rational choice of design variable which can be made to fit various prescribed design requirements and at the same time account for known changes in the function g by simply varying the probability which according to Equation 4 also implies a change in t_{0D}.

The analysis now proceeds as follows: Using assumption I, a probability distribution F is defined on t_0 where t_0 is the set of lowest annual daily average temperatures. The probability distributions for a sample of cities for which design

[1] Exponent numerals refer to References.

temperatures are desired (and for which the present design temperatures are known) are then fitted for each city of the sample. From this set of distributions a single probability is chosen such that the t_{0D} determined by the F's are equal on the average to the present design temperatures[1]. This is chosen as the central design value to which are added 4 temperatures: 2 with probabilities below the central value and 2 with probabilities above. This produces a set of 5 design values with varying probabilities. The central values are equal on the average to present design values and are homogeneous with respect to probability, as will be seen later. The additional 4 values allow the engineer freedom in choosing a design value better

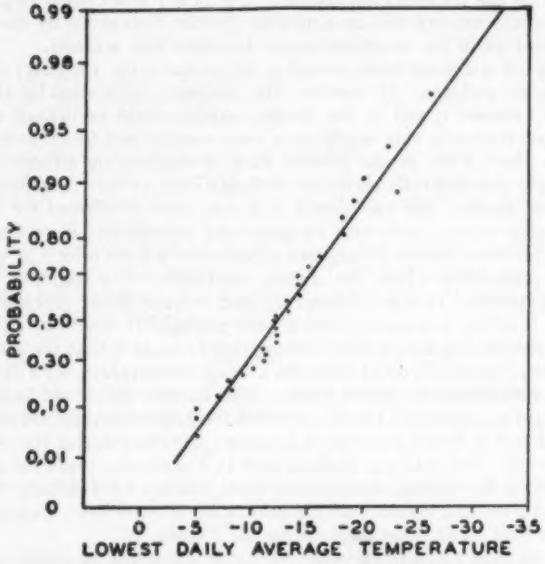

FIG. 1—EXAMPLE OF METHOD USED IN FITTING A FREQUENCY CURVE FROM EQUATION 8 TO 30-YEAR RECORDS (CURVE IS FOR MINNEAPOLIS)

suited to his knowledge of g and prescribed design requirements, by moving up or down from the central value with which he has had long experience. Should he have no special detailed knowledge about g or his prescribed design requirements, he may still use the central value with assurance that he is not departing from present design practice[1] except in the instances where present values are excessively biased.

CLIMATOLOGICAL ANALYSIS

The problem of climatological analysis is to determine the function F of Equation 4. This consists in fitting a theoretically suitable frequency curve to the

t_0 for each city for which design data are required. The t_0 are extreme (or minimum) annual values which are a sample from a much larger set of temperatures, *i.e.* each year has 365 daily averages from which one minimum is chosen. It is reasonable to expect, therefore, that the Type I extreme value distribution theory will apply[3]. This distribution is given by the equation

$$F(t_0) = e^{-e^{-[(\alpha - t_0)/\beta]}} \qquad \text{(8)}$$

where

$F(t_0)$ = the probability of $t < t_0$ (see Equation 4).
e = the base of natural logarithms.
α and β = parameters of the distribution.

The statistical problem consists in estimating satisfactory values of α and β from a series of values of t_0. Lieblein[3] has provided a method of fitting Equation 8 which was found to be very satisfactory for this purpose. Since the method of fitting is discussed in detail by Lieblein, it is not necessary to discuss it here. It may be said, however, that the method is very simple, although it should be mentioned that Lieblein only discusses the fitting of maxima so an adjustment must be made to his methods for fitting to minima as required here.

In preparing the design values presented here, the distribution of Equation 8 was first fitted to 30-year records ending in 1950 (standard normal period) for a well distributed climatic sample of 23 cities.* An example of the fit obtained is shown for Minneapolis in Fig. 1. Here the empirical probabilities determined from the samples are plotted as points on a special probability paper which make data which are distributed as Equation 8 plot on a straight line. The line shown in Fig. 1 is the theoretical line obtained from the values of α and β estimated by Lieblein's method. It is seen that the empirical points fit the line very well. This was found to be the case in all 23 cities analyzed in detail.

It should be noted that the probabilities given on the ordinate of Fig. 1 are the probabilities of an annual lowest daily average temperature being greater than a value read on the abscissa. For the design problem probabilities of being lower than a particular value are required; hence the complements of the probabilities of Fig. 1 are needed. These are given by simply subtracting those on the ordinate from one. For example, reading from Fig. 1, the temperature corresponding to a probability of 0.95 is −25 F. Subtracting 0.95 from one gives 0.05 which corresponds to the value given for Minneapolis in Table 1.

PROPOSED WINTER DESIGN DATA

Using assumption II proposed in the discussion of the design variable, a value of probability on t_0 was found for the 23 sample stations such that the present design temperature t_d is equal on the average to a new t_0. This is labelled $t_{0.075}$ in Fig. 2 where the points for the 23 cities are plotted. The line indicates the equality of the 2 design temperatures and averages the relationship within the limits of sampling variability and bias in the present design data t_d. The circled points are those for which the $t_{0.975}$ given by THE GUIDE[1] are also available. The probability on t_0 which made t_0 equal to t_d on the average is 0.075. This is equivalent to an

* These 23 cities are designated with asterisks in Table 1.

occurrence of about once in 13 years which seems to be a reasonable estimate of the probability of a system's failure to meet a load using the presently available design data. This was therefore adopted as the central value for t_{0D} not only because of its reasonable probability but because of its relation to t_d on which a long experience in heating system design has accumulated.

Four other values of probability were then chosen to bracket the $t_{0.075}$ and give flexibility along the design variable scale. This allows for other design requirements and the use of greater knowledge of the function g under specific circumstances. Two greater probabilities were chosen, 0.10 and 0.20, and 2 lesser probabilities, 0.05 and 0.025. This seemed to be a reasonable selection which was made

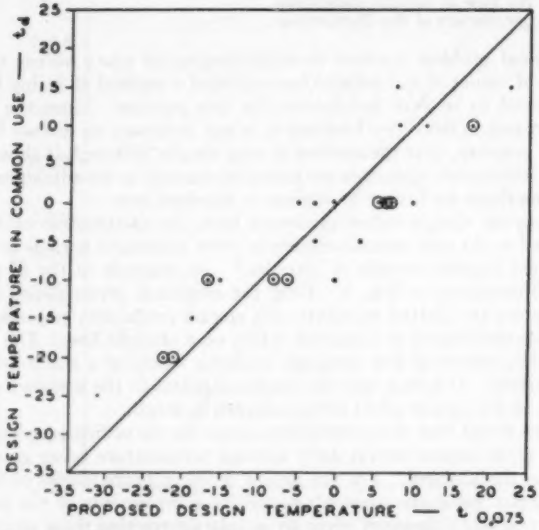

FIG. 2—RELATION OF DESIGN TEMPERATURE IN COMMON USE, t_d, AND PROPOSED OUTSIDE TEMPERATURE BASED ON 0.075 PROBABILITY, $t_{0.075}$, FOR 23 CITIES

in cooperation with the TAC on Weather Data. (Actually there is no real limitation on the number of probabilities obtainable since the distributions are available for all cities in Table 1). These then gave rise to the associated t_{0D}: $t_{0.20}$, $t_{0.10}$, $t_{0.05}$, and $t_{0.025}$ by the inverse of Equation 4, giving chances of one in 5, 10, 20, and 40 years. This set of values is given in Table 1 of the proposed design temperatures together with present design temperatures which may be compared with the central value $t_{0D} = t_{0.075}$. The difference $t_d - t_{0.075}$ is also given for ready comparison. The latter average close to zero for the entire table, showing that $t_{0.075}$ is equal on the average to t_d.

There are a number of striking examples of bias in the present design temperatures. For example, Table 1 shows the same design temperature as in THE GUIDE for Washington, Philadelphia, New York, and Boston, all zero. This appears

TABLE 1—PROPOSED OUTSIDE DESIGN TEMPERATURE DATA, t_{0D}, FOR VARIOUS RISKS BASED ON THE DISTRIBUTION OF THE LOWEST DAILY AVERAGE TEMPERATURE

(Cities designated with asterisks indicate those for which data are plotted in Figs. 2, 3, and 4)

COL. (1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
STATE	STATION	PROBABILITY OF LOWER TEMPERATURE					GUIDE DESIGN, t_d	t_d MINUS t_{0D} AT 0.075[a]
		0.025	0.05	0.075	0.10	0.20		
		$t_{0.025}$	$t_{0.05}$	$t_{0.075}$	$t_{0.10}$	$t_{0.20}$		
ALA.	Anniston	5	9	12	14	18	5	−7
	Birmingham	6	10	12	14	18	10	−2
	Mobile	17	20	22	24	27	15	−7
	Montgomery	11	15	18	19	23	10	−8
ARIZ.	Flagstaff	−12	−7	−4	−2	3	−10	−6
	Phoenix	33	35	36	37	39	25	−11
	Yuma	34	36	38	39	41	30	−8
ARK.	Bentonville	−8	−4	−1	0	5	—	—
	Fort Smith	4	7	9	11	14	10	1
	Little Rock	6	10	12	13	16	5	−7
CALIF.	Eureka	28	30	32	33	35	30	−2
	Fresno	29	31	32	32	34	25	−7
	Los Angeles	37	39	41	41	44	35	−6
	Sacramento	27	29	30	31	33	30	0
	San Diego	41	42	43	44	46	35	−8
	San Francisco	34	36	37	38	40	35	−2
COLO.	Denver	−20	−15	−12	−10	−5	−10	2
	Durango	−17	−11	−8	−6	0	—	—
	Grand Junction	−11	−6	−3	−1	4	−15	−12
	Leadville	−15	−12	−10	−9	−5	—	—
	Pueblo	−24	−18	−14	−12	−5	−20	−6
CONN.	Bridgeport	−6	−3	−1	1	4	—	—
	Cream Hill	−12	−9	−7	−6	−2	—	—
	Hartford	−7	−4	−2	0	3	0	2
	New Haven	−5	−2	0	1	5	0	0
	New London	−2	1	3	4	7	—	—
	Norwalk	−7	−3	−1	0	4	—	—
D. C.	Washington*	5	8	10	11	14	0	−10
FLA.	Jacksonville	24	27	28	30	33	25	−3
	Key West	50	52	53	54	56	45	−8
	Pensacola	19	22	24	25	29	20	−4
	Tampa	32	35	36	38	41	30	−6
GA.	Atlanta*	5	9	11	13	17	10	−1
	Augusta	15	18	20	21	25	10	−10
	Macon	14	18	20	21	24	15	−5
	Savannah	19	22	24	25	28	20	−4
IDAHO	Boise	−20	−14	−10	−7	0	−10	0
	Lewiston	−24	−16	−12	−8	0	5	17
	Pocatello	−27	−20	−17	−14	−7	−5	12
ILL.	Cairo	−8	−3	0	2	7	0	0
	Chicago	−17	−13	−11	−9	−5	−10	1
	Peoria	−20	−15	−13	−11	−6	−10	3
IND.	Evansville	−12	−7	−4	−2	3	—	—
	Fort Wayne	−13	−9	−7	−5	−2	−10	−3
	Indianapolis	−14	−10	−8	−6	−1	−10	−2
	Lafayette	−15	−11	−8	−6	−1	—	—
	Terre Haute	−12	−8	−6	−4	0	—	—
IOWA	Charles City	−28	−24	−21	−20	−16	—	—
	Davenport	−18	−14	−12	−10	−5	−15	−3
	Des Moines	−19	−15	−13	−12	−8	−15	−2
	Dubuque	−22	−18	−15	−14	−9	−20	−5
	Keokuk	−21	−16	−13	−11	−6	—	—
	Sioux City	−22	−18	−16	−15	−11	−20	−4
KAN.	Concordia	−17	−13	−11	−9	−5	−10	1
	Dodge City	−16	−11	−9	−7	−2	−10	−1
	Iola	−15	−10	−7	−5	0	—	—
	Topeka	−15	−11	−8	−7	−2	−10	−2
	Wichita	−13	−9	−6	−4	0	−10	−4
KY.	Bowling Green	−7	−1	2	5	11	—	—
	Hopkinsville	−7	−1	2	4	10	—	—
	Irvington	−8	−3	0	3	8	—	—
	Lexington	−9	−5	−2	−1	4	—	—
	Louisville	−9	−5	−2	0	4	0	2
	Shelbyville	−9	−4	−1	1	7	—	—

[a] Col. 8 minus Col. 5.

TABLE 1 (Continued)—PROPOSED OUTSIDE DESIGN TEMPERATURE DATA, t_{OD}, FOR VARIOUS RISKS BASED ON THE DISTRIBUTION OF THE LOWEST DAILY AVERAGE TEMPERATURE

COL. (1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
STATE	STATION	PROBABILITY OF LOWER TEMPERATURE					GUIDE DESIGN, t_d	t_d MINUS t_{OD} AT 0.075[a]
		0.025	0.05	0.075	0.10	0.20		
		$t_{0.025}$	$t_{0.05}$	$t_{0.075}$	$t_{0.10}$	$t_{0.20}$		
LA.	New Orleans	21	24	26	27	30	20	−6
	Shreveport	8	12	14	16	20	20	6
ME.	Eastport	−14	−11	−9	−8	−5	−10	−1
	Portland	−15	−11	−9	−7	−3	−5	4
MD.	Baltimore	3	6	8	9	13	0	−8
MASS.	Boston*	−6	−2	0	1	4	0	0
MICH.	Alpena	−16	−12	−10	−8	−5	−10	0
	Detroit	−10	−6	−4	−3	1	−10	−6
	Escanaba	−25	−21	−18	−17	−12	−15	3
	Grand Haven	−8	−5	−3	−1	2	—	—
	Grand Rapids	−9	−6	−4	−2	1	−10	−6
	Houghton	−24	−20	−18	−16	−11	—	—
	Lansing	−14	−10	−8	−7	−3	−10	−2
	Ludington	−13	−9	−7	−5	−1	—	—
	Marquette	−22	−18	−16	−14	−10	−10	6
	Sault Ste. Marie	−24	−21	−19	−17	−14	−20	−1
MINN.	Duluth	−32	−29	−27	−25	−22	−25	2
	Minneapolis*	−28	−25	−23	−21	−18	−20	3
	Moorehead	−35	−32	−29	−27	−23	—	—
	St. Paul	−28	−25	−23	−21	−18	−20	3
MISS.	Corinth	−2	4	8	11	18	—	—
	Meridian	8	12	14	16	20	10	−4
	Vicksburg	9	13	15	17	21	10	−5
MO.	Columbia*	−16	−11	−9	−7	−2	−10	−1
	Hannibal	−19	−14	−12	−9	−4	—	—
	Kansas City	−15	−11	−8	−6	−2	−10	−2
	St. Louis*	−12	−8	−5	−3	1	0	5
	Springfield	−12	−8	−5	−4	1	—	—
MONT.	Billings*	−42	−35	−31	−28	−21	−25	6
	Havre	−49	−43	−39	−36	−29	−30	9
	Helena	−53	−44	−39	−36	−27	−20	19
	Kalispell	−43	−35	−31	−27	−19	−20	11
	Miles City	−44	−38	−35	−32	−26	−35	0
NEB.	Lincoln*	−22	−18	−15	−14	−9	−10	5
	North Platte	−21	−17	−15	−13	−9	−20	−5
	Omaha	−24	−19	−17	−15	−10	−10	7
	Valentine	−27	−23	−21	−19	−15	−25	−4
NEV.	Reno*	−4	0	3	5	9	−5	−8
	Winnemucca	−17	−12	−9	−6	0	−15	−6
N. H.	Concord	−17	−13	−11	−9	−5	−15	−4
N. J.	Atlantic City	3	6	8	9	12	5	−3
	Elizabeth	−4	1	3	5	10	—	—
	Flemington	−5	0	3	5	10	—	—
	New Brunswick	−1	3	5	7	11	—	—
	Plainfield	−2	2	4	6	10	—	—
	Somerville	−6	−2	1	3	8	—	—
	Trenton	−4	0	2	3	7	0	−2
N. M.	Albuquerque	3	6	8	10	14	0	−8
	Roswell	−4	1	4	6	11	−10	−14
	Santa Fe	−5	0	2	4	9	—	—
N. Y.	Albany	−16	−12	−9	−8	−4	−10	−1
	Binghamton*	−12	−9	−7	−5	−2	−10	−3
	Buffalo	−11	−7	−5	−4	0	−5	0
	Canton	−28	−25	−22	−21	−17	−25	−3
	Ithaca	−9	−6	−4	−3	0	−15	−11
	New York*	1	3	5	6	9	0	−5
	Oswego	−12	−9	−7	−5	−1	−10	−3
	Rochester	−9	−6	−4	−3	1	−5	−1
	Syracuse	−16	−12	−10	−8	−4	−10	0
N. C.	Asheville	−2	2	5	6	11	0	−5
	Charlotte	9	12	14	16	19	10	−4
	Hatteras	16	19	21	22	25	—	—
	Raleigh	9	12	14	15	18	10	−4
	Wilmington	16	19	20	22	25	15	−5

TABLE 1 (Continued)—PROPOSED OUTSIDE DESIGN TEMPERATURE DATA, t_{OD} FOR VARIOUS RISKS BASED ON THE DISTRIBUTION OF THE LOWEST DAILY AVERAGE TEMPERATURE

Col. (1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
State	Station	Probability of Lower Temperature					Guide Design, t_d	t_d Minus t_{OD} at 0.075[a]
		0.025	0.05	0.075	0.10	0.20		
		$t_{0.025}$	$t_{0.05}$	$t_{0.075}$	$t_{0.10}$	$t_{0.20}$		
N. D.	Bismarck	−38	−34	−31	−29	−24	−30	1
	Devils Lake	−37	−34	−32	−30	−27	−30	2
	Grand Forks	−33	−29	−26	−24	−20	—	—
	Williston	−44	−38	−35	−33	−27	−35	0
Ohio	Cincinnati	−9	−5	−3	−1	3	0	3
	Cleveland	−11	−7	−5	−3	1	0	5
	Columbus	−9	−5	−3	−2	2	−10	−7
	Dayton	−10	−6	−4	−2	2	0	4
	Sandusky	−10	−6	−4	−2	2	0	4
	Toledo	−11	−7	−5	−4	0	−10	−5
Okla.	Oklahoma City	−9	−4	−1	1	6	0	1
Ore.	Baker	−24	−17	−14	−11	−5	−5	9
	Portland	1	7	10	12	18	10	0
	Roseburg	14	17	19	20	24	10	−9
Pa.	Erie*	−9	−5	−3	−2	2	−5	−2
	Harrisburg	0	2	4	5	8	0	−4
	Philadelphia*	1	4	6	7	11	0	−6
	Pittsburgh*	−9	−6	−3	−2	2	0	3
	Reading	−2	1	3	4	8	0	−3
	Scranton	−8	−4	−2	−1	3	−5	−3
R. I.	Block Island*	3	5	7	8	10	0	−7
	Kingston	−5	−2	0	1	4	—	—
	Providence	−4	−1	1	2	5	0	−1
S. C.	Charleston*	17	20	22	23	27	15	−7
	Columbia	14	17	19	20	24	10	−9
S. D.	Huron*	−27	−23	−21	−20	−17	−20	1
	Pierre	−22	−19	−17	−15	−12	—	—
	Rapid City	−28	−24	−22	−20	−16	−20	2
Tenn.	Chattanooga*	0	5	8	10	14	10	2
	Knoxville	−3	2	5	7	12	0	−5
	Memphis	0	4	6	8	13	0	−6
	Nashville	−4	0	3	5	10	0	−3
Tex.	Abilene*	0	5	7	9	14	15	8
	Amarillo	−10	−5	−2	0	5	−10	−8
	Corpus Christi	16	20	23	24	29	20	−3
	Dallas	1	6	8	10	15	0	−8
	El Paso	15	18	20	21	25	15	−5
	Fort Worth	0	5	8	10	15	10	2
	Galveston	16	20	23	24	29	20	−3
	Houston	13	17	19	21	25	20	1
	Palestine	4	8	11	13	18	15	4
	San Antonio	12	16	19	20	25	20	1
	Taylor	5	9	12	14	19	—	—
Utah	Salt Lake City*	−8	−3	−1	1	6	−10	−9
Vt.	Burlington*	−23	−19	−17	−15	−11	−10	7
	Northfield	−24	−21	−19	−17	−13	—	—
Va.	Cape Henry*	13	16	17	18	21	10	−7
	Lynchburg	7	10	11	12	15	5	−6
	Norfolk	11	14	15	17	20	15	0
	Richmond	6	9	11	12	16	15	4
	Wytheville	−3	1	3	4	8	—	—
Wash.	Seattle	9	13	15	17	21	15	0
	Spokane	−28	−20	−16	−13	−5	−15	1
	Tacoma	9	13	15	16	20	15	0
	Tatoosh Island	12	16	18	20	25	15	−3
	Walla Walla	−23	−16	−12	−8	−1	—	—
W. V.	Elkins	−10	−7	−4	−3	1	−10	−6
	Parkersburg	−7	−3	−1	1	5	−10	−9
Wis.	Green Bay	−27	−23	−20	−18	−14	−20	0
	La Crosse	−27	−23	−20	−19	−14	−25	−5
	Madison	−25	−21	−19	−17	−13	−15	4
	Milwaukee	−24	−20	−17	−15	−10	−15	2
Wyo.	Cheyenne*	−26	−22	−19	−17	−12	−15	4
	Lander	−39	−33	−30	−27	−21	−18	12
	Yellowstone Park	−45	−38	−34	−30	−23	—	—

strange when the variation in climate among these 4 cities is considered. The proposed central design values place these in proper climatic order: Washington 10 F, a value reported to be often used, Philadelphia 6 F, New York 5 F, and Boston 0 F. Boston remains the same as the present value in THE GUIDE. Other homogenization may be noted by those interested in design for particular areas.

Two other comparisons were made with published design values. In Figs. 3 and 4 the $t_{0.975}$ as given in THE GUIDE for the cities corresponding to the circled points (Fig. 2) are compared with the central design value $t_{0.075}$ and the design value in common use t_d. (Note that $t_{0.975}$ is $t_{0.025}$ in the nomenclature employed here since probabilities are for failure to meet load, or 1 − 0.975. This is *not* equal

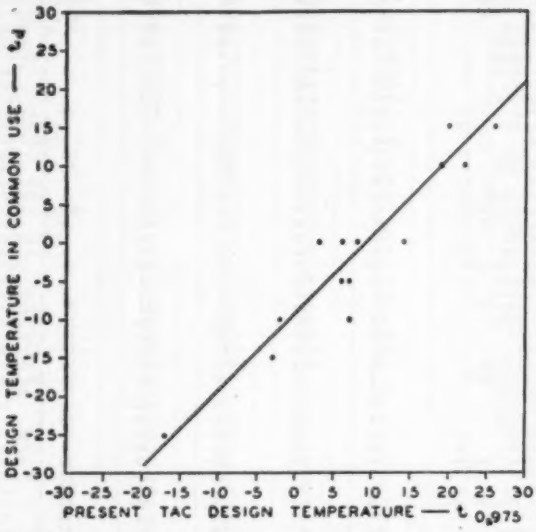

FIG. 3—RELATION OF DESIGN TEMPERATURE IN COMMON USE, t_d, AND PRESENT TAC DESIGN TEMPERATURE, $t_{0.975}$, BASED ON 0.025 PROBABILITY FOR 14 CITIES

to $t_{0.025}$ of the proposed design values). The comparisons show why the $t_{0.975}$ has not been very popular, for it gives design values about 10 deg higher on the average than both the values in common use and the proposed central values, yet it has a lower probability, 0.025 as compared to 0.075. To obtain values on this design variable comparable to present practice would require lowering the probability to values which are inconsistent with reasonable risk or contingency for heating system design. These lower probabilities would also be difficult to estimate accurately from available data.

It will be recalled that the $t_{0.975}$ is the temperature for which the average proportion of hours above $t_{0.975}$ is 0.975 during the winter season December, January, and February. This may still have uses, which up to the present time have not been

developed. As seen in Figs. 3 and 4, it may be obtained from a relationship established with either $t_{0.075}$ or t_d. This is the subject of a study by John Everetts, Jr., Philadelphia, Pa., who has given it much more thorough consideration.

Although this discussion has meant to carefully avoid discussing load problems except for pointing out difficulties which affect the choice of a basis for design temperatures, 2 suggestions on the use of data in the proposed form might be helpful. In cases where the designer has a limited knowledge of g and the prescribed design requirements, for example, in a structure which could be later used for unanticipated purposes, a smaller probability within economic restrictions could be considered. A

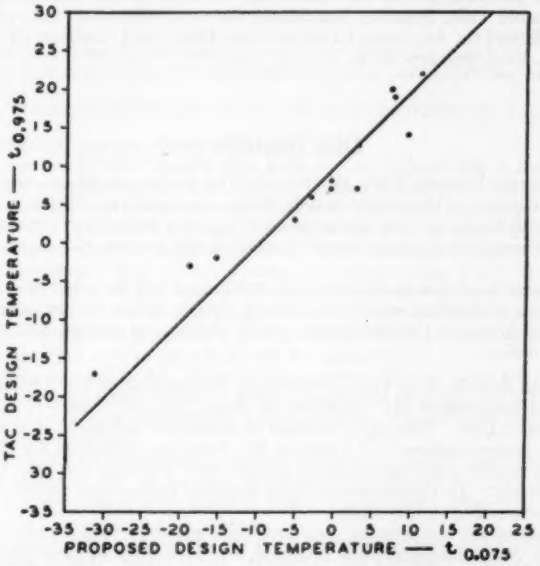

FIG. 4—COMPARISON OF PRESENT TAC DESIGN TEMPERATURE AND PROPOSED DESIGN TEMPERATURE FOR 13 CITIES

second instance might be to differentiate the use of the structure by the probabilities. For example, a hospital might naturally call for a smaller probability of failure to meet load than an office building. Many other applications of the flexibility provided by a range of design values, in contrast to a single value, would be found by the designer as experience in use is developed.

ACKNOWLEDGMENTS

The author wishes to express appreciation for many constructive discussions with members of the TAC on Weather Data and more especially for the helpful sug-

gestions and interest of Mr. Everetts, the chairman, Scott Heider, staff engineer, Building Research Advisory Board, and P. R. Achenbach, chairman of the TAC on Heating Load. Sincere appreciation must also go to Daniel Harton of National Weather Records Center, Asheville, and to my statistical assistant, Maurice Kasinoff, for carrying out all computations.

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DISCUSSION

CARL W. SIGNOR, Detroit, Mich. (WRITTEN): The author should be commended for his analysis and study of the winter outside design temperature. He has removed the limitation of THE GUIDE outside design temperature and has given architects and engineers a new expression of design temperatures and the possible risk involved in their use.

When this table is applied in the design of buildings it will be found that for each of the various types of building structure a definite outside design temperature should be used. This should reduce the installation cost of the heating systems and still provide the required comfort.

J. B. GRAHAM, Buffalo, New York, (WRITTEN): As a member of the TAC on Weather Data I wish to compliment Mr. Thom on his paper, both as to subject material and method of presentation. This paper is based on statistical techniques that may not be familiar to the design engineer, but I believe Mr. Thom has organized his material in a fashion that is readily understandable to anyone who will take the time to give this paper serious study. This paper also includes excellent background information on the various parameters used in this study.

While a study of the temperature data presented will show that for most cities very little change in design temperature is proposed, I believe there is, nevertheless, a great deal of material in this paper that will interest the design engineer. The sources of previous design temperatures were, in many cases, obscure, but Mr. Thom has presented a rational method of computing outside winter design temperatures. The design engineer will feel greater confidence in these temperatures based, as they are, on a statistical analysis of weather data. This approach should be especially useful in economic studies and evaluations.

There is one point that may not be clear and I mention it here because it was the subject of some discussion at the TAC meeting. It will be noticed that these data are based on the lowest *average* daily temperatures. Daily low temperatures were not used because the erratic nature of extreme temperatures would have resulted in unreasonable probabilities. These unreasonable probabilities result because there is much greater probability that daily low temperatures will drop below a selected design temperature than would be the case in using daily averages. These computations based on average temperature eliminate the effect of extreme values that would affect the final result, but would not reflect the true probability.

As Mr. Thom has noted, this paper covers the contingencies in the design of heating systems that arise from the variability of the weather. By a climatological analysis this

contingency has been reduced to a probability curve. Thus the engineer has an advantage of design temperature flexibility not previously available to him. It is to be hoped that work along this general line will be continued by the Society and that this advantage will soon be available in other areas of weather data.

JOHN EVERETTS, JR., Philadelphia, Penn., (WRITTEN): On behalf of TAC on Weather Data, I wish to compliment Mr. Thom on his paper and to thank him for his untiring efforts which culminated in winter data which this Society has been requesting for many years.

The data presented in his paper are basic, easily reproducible from Weather Bureau publication data, and cover all the years on record for the particular station. The validity of these data is beyond question.

When these data were first presented to the TAC, several questions arose as to their adaptability and usefulness in design problems.

First, how did those figures compare with that in common use?

Secondly, how could it be related in terms of hours per month, instead of frequency in years?

Thirdly, of what significance do the 5 columns of probability have in winter design?

These questions are not simple to answer.

The comparison of Mr. Thom's data with that in common use is shown in the last column of his tabulated data. This column shows the difference between the common use values now in THE GUIDE and the temperature corresponding to the 0.075 probability. Of the 153 cities in THE GUIDE, only 50 show more than a 5 deg difference and only 9 show more than a 10 deg difference. This is not particularly significant since Mr. Thom has shown that the 0.075 value for Boston, New York, Philadelphia, and Washington, D. C. is 0, 5, 6, and 10 F respectively, whereas THE GUIDE data shows 0 F for all these cities even though Washington, D. C. is some 500 miles south of Boston. This would indicate that the present GUIDE values which are based upon limited data, are neither accurate nor logical and should be corrected.

The question of relating Mr. Thom's data to some value of hours per month requires an analysis on weather data which is of such magnitude that the cost is far beyond that which can be provided by the Society, and apparently is not of sufficient importance to interest industry support.

A spot check of 8 cities selected at random indicates a possible connection between the probability values of Mr. Thom and the hours for the month of January 1935 to 1939 inclusive. I would like to show 2 figures (Figs. A and B) giving the data for the 8 cities studied.

The solid line in each case shows the percentage probability and corresponding temperatures given in Mr. Thom's paper. The dash line indicates 10 times the percentage of the number of hours in January at which the temperature is at or below that indicated. For example, the 20 percent probability for Atlanta (Fig. B) shows a temperature of 17 F. Two percent of the total hours in January shows a temperature which is equal or lower than 18 percent. Similarly, Boston (Fig. B) shows a temperature of 2 F at the 15 percent probability and also a temperature which is equal to or lower than 2 F for the month of January for 1.5 percent of the time.

Similar data is shown for Buffalo, N. Y.: Detroit, Mich.; Memphis, Tenn.; Washington, D. C.; Philadelphia, Pa.; and New Orleans, La.

This discussion does not propose, at this time, to prove a relationship between probability values of Mr. Thom and the hourly data for the month of January. The relationship indicated by these curves is shown as a matter of interest and it is believed that their close proximity and shape are more than just coincidence. Further studies will be continued along these lines.

The 5 probability values for each city (which can be plotted to obtain any desired intermediate value between 0.025 and 0.20) is to give the design engineer the choice based upon his judgement of a design temperature for a specific problem.

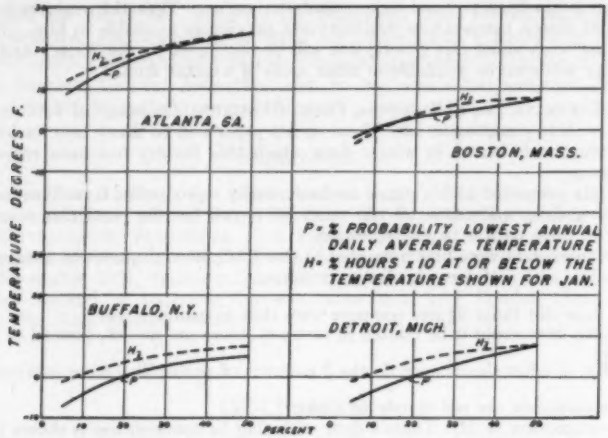
TEMPERATURE DEGREES F.
ATLANTA, GA.
BOSTON, MASS.
BUFFALO, N.Y.
DETROIT, MICH.
P = % PROBABILITY LOWEST ANNUAL DAILY AVERAGE TEMPERATURE
H = % HOURS x 10 AT OR BELOW THE TEMPERATURE SHOWN FOR JAN.
PERCENT

FIG. A—RELATION BETWEEN AUTHORS PROBABILITY (P) AND PERCENT HOURS × 10 AT OR BELOW TEMPERATURE SHOWN FOR JANUARY (H), AS PLOTTED FOR FOUR U. S. CITIES

For example; a hospital of light construction where 24 hour control of temperature is necessary, would require a lower design temperature than an office building of heavy construction which is occupied during the day only. The actual design temperature is the prerogative of the design engineer. The probability values shown in Mr. Thom's

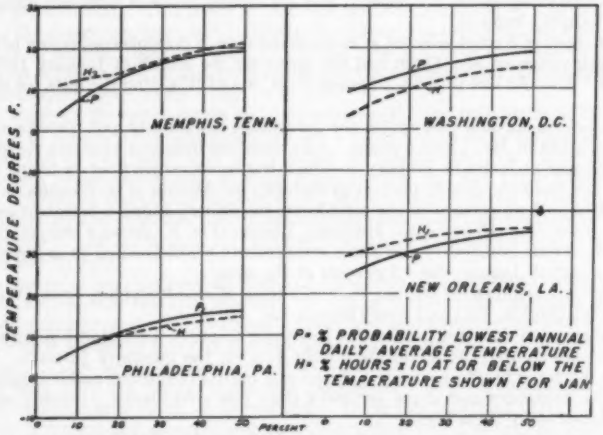
TEMPERATURE DEGREES F.
MEMPHIS, TENN.
WASHINGTON, D.C.
NEW ORLEANS, LA.
PHILADELPHIA, PA.
P = % PROBABILITY LOWEST ANNUAL DAILY AVERAGE TEMPERATURE
H = % HOURS x 10 AT OR BELOW THE TEMPERATURE SHOWN FOR JAN
PERCENT

FIG. B—SAME RELATION AS IN FIG. A BUT APPLIED TO FOUR OTHER TYPICAL U. S. CITIES

paper are presented merely as a guide to show the order of magnitude of deviation above and below the 0.075 value.

This is the first attempt at presenting weather data in this unique form and while the TAC feels it is the most accurate and most complete weather data for Society dollars expended, we would like the expression as to its value from those members of the Society who make use of such data.

W. R. BRISKEN†, Bloomfield, N. J., (WRITTEN): Calculations of heating loads for buildings are subject to a great number of limiting assumptions which are generally connected with 3 different sets of characteristic values; they are weather, building structure and internal loads. Inasmuch as it is and always will be impossible (for reasons introduced by nature and resulting from practical considerations) to obtain ultimate accuracy in load calculations, it is important to analyze the margin of inaccuracy introduced by the method of calculation and by the selection of variables.

Mr. Thom is to be congratulated for the contribution which he made with his paper. By providing 5 design temperatures with different probabilities that they may be exceeded he makes it possible for the design or application engineer to determine in advance the effect of one variable on the over-all result, i.e. on inside temperature. The engineer can now appraise the risk which he is taking and does not need guessing on the effect of that variable.

The first simplifying assumption especially attracted my curiosity. The question arose if actually the *greatest stress on the heating system will occur when the average temperature, for a day or a linear function of it, assumes its lowest value.* This assumption appears definitely justified for buildings with large thermal time constants. However, most one-family houses are of lighter construction with time constants of around 4 to 8 hours. In these cases the lowest hourly temperature may affect the heating load more than the lowest daily average temperature. A limited amount of weather records (procured from the U. S. Weather Records Center) was available from an earlier study. So I checked if perhaps the lowest daily average temperature and the lowest hourly temperature would occur on the same day. This actually was the case for the stations and years checked: New York, N. Y. 1950 and 1952; Charleston, South Carolina 1948; and Madison, Wisconsin 1951. It would be interesting to hear from Mr. Thom what the probability is that the 2 minimum temperatures occur on the same day. It was also noted that on days with less severe temperatures the prevailing winds would not have affected *assumption I* adversely. However, again it would be very valuable to have a larger statistical sample to prove this point.

If it can be proven that *assumption I* is valid for all locations of interest then design temperatures with probabilities as recommended by Mr. Thom and supplemented with wind data would be most suitable to people using the steady state load calculating method. For those who believe in the necessity of applying transient load calculating methods, it would be desirable to have in addition, temperature slopes leading to and following the design temperatures. Reviewing the weather data mentioned before, the average gradients in deg F per hour listed in Table A were found:

TABLE A

	6 HRS. BEFORE	6 HRS. AFTER	12 HRS. BEFORE	12 HRS. AFTER
New York, N. Y.	0.0	0.0	0.2	0.8
Charleston, S. C.	1.3	2.0	1.25	0.75
Madison, Wisconsin	1.2	4.0	1.5	2.3

† Manager, Advance Engineering, General Electric Company.

It is believed that the error resulting from the use of average slopes will not affect the final result unduly. However, the margin of possible error requires engineering evaluation. It would be rather simple to add the information on temperature slopes to the design temperatures and thereby to make the data of design temperature even more valuable for refined load calculations.

I like to express my sincere hope that Mr. Thom will continue his valuable work, in order to provide all design weather data needed for heating and cooling load calculations with probabilities. Since in the search for maximum load conditions weather and building structure cannot be divorced entirely it may be necessary to appraise *apparently* extreme weather conditions for their effect on thermal loads of buildings of different structure. But whatever the final presentation of design weather data may look like the single factor will stand out that probability information will make the data considerably more valuable for practical use.

P. R. ACHENBACH, Washington, D. C., (WRITTEN): Mr. Thom is due a vote of thanks for providing revised winter outside design temperatures based on a climatological analysis of 30 years of weather data. I believe some or all of the data should be used in THE GUIDE to replace the present winter design temperatures because the values presented in the paper are internally consistent and based completely on records of the Weather Bureau, whereas the values now published were collected from various sources.

There appears to be one major problem in publishing design temperatures based on 5 different probability values as suggested in this paper. The choice may be too great and the guidance that can be furnished in choosing among them too little to serve as a satisfactory basis for fair competition among heating contractors all striving for the same contracts. I believe the Guide should carry a strong recommendation toward the use of a particular probability column, say the 0.075 probability for general application where special conditions are not known to exist.

The analysis is based on the minimum daily average temperature each year for 30 years. This means that on such a day the outdoor temperature will be below the average value approximately 12 hours, probably during the night. Buildings of low heat capacity would tend to respond to the diurnal temperature cycle and might become underheated during the low portion of the diurnal cycle.

Some of the construction features, occupancies, heating system characteristics, and climatic variables that would suggest the choice of lower design temperature values are as follows:

1. Walls of low heat capacity
2. Large amounts of glass in the exterior walls
3. Excessive air infiltration
4. High ventilation rates
5. Fixed output heating systems such as resistance heaters
6. A heating system whose capacity decreased at lower outdoor air temperatures such as an air-to-air heat pump
7. Hospitals and old folks homes
8. Areas where the diurnal temperature cycle is of wide amplitude

On the other hand, building characteristics, usage and climatic factors favoring the choice of higher outside design temperatures are:

1. Walls of high heat capacity
2. Heavily insulated constructions
3. Buildings requiring comfort in the daytime only
4. Heating systems with flexible heat output
5. Areas with small diurnal temperature change.

A study of Table 1 shows that the difference in the daily average outside temperature that occurs once in 5 years and that which occurs once in 40 years is of the order of 10

deg in a great many cities, but becomes as large as 20 deg or more in cold areas such as Montana or in Yellowstone Park. This reveals that the probabilities corresponding to the *97½ percent value* and the *design value in common use* now published in THE GUIDE are widely different because these 2 outside design temperatures differ by 10 deg in many cases.

Climatological data could be analyzed on other bases than that used by the author and some of these might be easier to use than the 5 values proposed in this paper. For example, the effect of the heat capacity of a structure on the selection of the appropriate outside design temperature is related to the duration of the *cold spells*. Thus, if it were known how many times a year, or preferably how many times in 30 years the outside temperature remained at or below selected values for 2 consecutive hours, 4 consecutive hours, 8 consecutive hours, and so on, probability values could be developed that would have application to structures of widely different heat capacities. On the other hand, if the diurnal temperature cycle at a given location could be mathematically expressed as a sine wave with certain harmonics, its effect on the heat loss of a structure could probably be related to the daily average temperature together with the daily amplitude of the cycle. I am told by the author that such analyses would be many times more costly than that undertaken for the present paper and, therefore, could not be done at this time.

F. H. FULLER‡, Wilmington, Del.: 1. Many industrial plants are located in suburban or rural areas. We have found differences of 2 to 3 deg F between design temperatures (both winter and summer) based upon city office and airport data. Are the design temperatures specified in the paper based upon city office or airport data?

2. Does the author intend to carry out the same sort of analyses for summer design temperatures?

G. S. CAMPBELL, Chattanooga, Tenn.: The data appear to be very complete and good. For our particular locality (Chattanooga, Tennessee) the present design temperature in common use as shown in THE GUIDE is plus 10 F and apparently the 0.075 median tends to agree with this figure. My experience indicates that the TAC 97½ percent basis giving a 19 F design is much more realistic. For instance, we are an electric heat region and heat loads based on plus 10 F design result in much larger equipment requirements than the electric utility recommends and yet the utility recommendations give satisfactory installations. This makes the 0.075 median appear too conservative. Does this 0.075 median give better correlation with actual results in other areas or will it result in larger heat loads and more equipment on the job than really needed?

AUTHOR'S CLOSURE: I wish to thank the discussers for bringing out a number of points which I had not handled adequately. As is well known, the problem of outside design temperatures is a complicated one and has been studied from many angles. It is quite easy to make the solution complex and too difficult to tabulate and apply.

I am especially appreciative for the remarks of Mr. Graham who, it is clear, understands thoroughly the problem of design data. His explanations have added considerably to the general understanding of the paper.

Mr. Everetts has put forward an extremely interesting and ingenious extension of our results. This is readily applied and should be of aid both as a bridge with hourly frequency analyses and as a quick method for obtaining operating data.

I also appreciate Mr. Brisken's remarks and suggestions, and I am very much in agreement with the refinements he proposes. Mr. Brisken, I am sure, understands as do others, that we had to begin with a simple revision of the long standing design data. It was apparent to us that even to propose 5 values for THE GUIDE in place of one was already straining the space allowances a bit.

Many of the previous remarks also apply to Mr. Achenbach's discussion and I am in full agreement with what he has had to say. In the future, we must continue to re-

‡ Engineering Department, E. I. DuPont de Nemours & Company.

fine our design data with a view to putting the design of heating systems on as rational and exact a basis as possible.

Mr. Signor has made a helpful remark with respect to the efficiency of design. If every engineer employs these data with the understanding he shows for their use, considerable savings in heating system design should accrue.

The question of Mr. Fuller may be answered by saying that the sources of these data are mainly downtown observation stations. It must be agreed that there are differences between downtown and suburban and rural data. This problem needs further study. We intend to make similar analyses for summer design conditions

Mr. Campbell has raised a very good point. I had raised a similar point in the TAC in connection with the discussion of possible instructions on the use of the design temperatures. There is little doubt that in southern locations the designer might find it desirable to use a higher probability value. This could be accomplished perhaps by using the 0.20 probability value.

1595

No. 1595

ANALYSIS OF AN AIR-CONDITIONING THERMAL CIRCUIT BY AN ELECTRONIC DIFFERENTIAL ANALYZER

By G. V. PARMELEE*, P. VANCE**, AND A. N. CERNY†

This paper is the result of research carried out by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio.

THE NEED for making more realistic estimates of the instantaneous loads on air-conditioning systems has encouraged the development of new methods of load estimating. Mainly, the problem is one of taking into account the complex inter-relationship that exists between the outdoor environment with its periodic changes, and the structure with its thermal resistances and capacitances and internal energy sources. The principles of thermal circuits and circuit diagramming have been developed to describe this complex relationship. Analysis of the circuit diagram provides the instantaneous system loads, room surface temperatures and other data needed by designers in sizing equipment and selecting controls.

Several methods have been used successfully to solve air-conditioning thermal circuits. Reference 1 describes circuit principles and a mathematical procedure in complete detail. Hydraulic analogue,[2,3] electric analogue,[4] and differential analyzer methods[5] have also been used recently. This paper demonstrates the use of a standard electronic differential analyzer in solving a practical problem and seeks to serve the following purposes: (1) To demonstrate the principles in setting up

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** Goodyear Aircraft Corp., Akron, Ohio, and now at Massachusetts Institute of Technology, Lincoln Laboratory, Cambridge, Mass.
† Research Mathematician, ASHAE Research Laboratory.
‡ Exponent numerals refer to References.
Presented at the 63rd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Chicago, February 1957.

problems for solution by this method and to thereby encourage engineers to make use of such equipment, which is coming rather rapidly into common use. (2) To explore more fully the suitability of the analyzer in solving a large number of typical thermal-system problems with the view of putting results in a form for general use by air-conditioning engineers. This would involve systematically varying the circuit parameters. (3) To provide results for a specific problem which

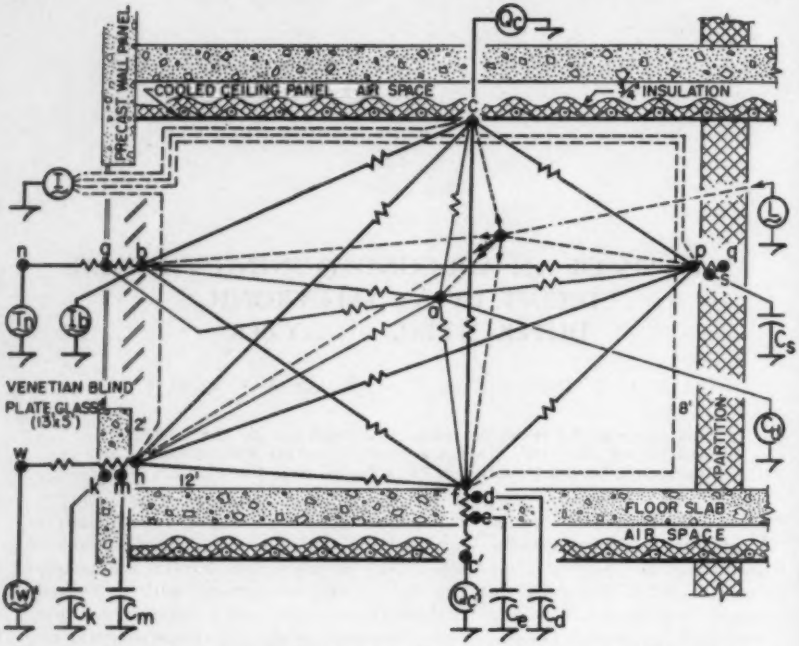

FIG. 1—SECTION THROUGH ROOM AND THERMAL CIRCUIT OF ILLUSTRATIVE PROBLEM. DIMENSIONS—12 x 25 x 8-FT CEILING

may be added to those of others in achieving a more thorough and practical understanding of thermal-system behavior.

ILLUSTRATIVE PROBLEM

The illustrative problem used as a means of studying the differential-analyzer technique is one of determining the time-variable loads on an office cooling system consisting of a cool-air supply system plus a cooled ceiling panel.

System loads arise from:

1. Solar radiation transmitted through a window provided with an indoor venetian blind.
2. The combined effect of sun and outdoor air in warming the window and blind.

3. The combined effect of sun and outdoor air in warming the exposed wall.
4. A steady lighting load for a portion of the 24-hr period.

Physical Aspects: The main physical features of the thermal system used to illustrate the differential-analyzer technique are shown in Fig. 1. The office to be conditioned is 12 x 25 ft with an 8-ft ceiling. Adjacent rooms, those above and below, and the corridor are held at the same temperature as the office under consideration. Only the southwest wall, which is 8 x 25 ft, is exposed. A 13-ft x 5-ft

TABLE 1—STRUCTURAL, THERMAL AND LIGHT REFLECTANCE DATA FOR ROOM COMPONENTS

COMPONENT	CIRCUIT JUNCTION SYMBOL	AREA SQ FT	MATERIAL	THICKNESS IN.	k BTU/(HR)(SQ FT)(F/FT)	w LB/CF	Cp BTU/(LB)(F)	LIGHT REFLECTANCE
Exposed Wall	h	135	light weight concrete plus plaster	4 overall	0.33	100	0.21	0.50
Floor	f	300	light weight concrete	4 overall	0.22	75	0.21	0.15
Ceiling	c	300	suspended, metal with attached cooling pipes	—	—	—	—	0.70
Sound Insulation	—	300	glass fiber laid on ceiling	¾	0.025	12	0.20	—
Partitions	p	392	Steel	20 gage	35	487	0.12	0.15
			and Rock Wool fill	3	0.0225	6	0.2000	—
Window	g	65	Plate glass—0.923 transmittance for sun at normal incidence					
Venetian Blind	b	65	White slats (metal) set at 45 deg Ratio slat width to spacing = 1.2					

window is centered in this wall and is provided with an inside venetian blind. The suspended metal ceiling has cooling coils attached to the upper side, and these are covered with ¾-in. blanket insulation. Interior partitions are of hollow metal construction filled with sound-absorbing material. Structural details, thermal properties, and the room surface reflectances for short-wave solar and light energy are listed in Table 1.

The Thermal Circuit: The thermal circuit is diagrammed on Fig. 1. Room air is represented by junction point a, and is postulated to be at a uniform temperature of 75 F. All surfaces are postulated as isothermal, and their temperatures are denoted as T_a, T_p, etc. Conditions of the problem are further described as

follows (with circuit thermal resistances, capacitances, thermal potential and thermal current inputs tabulated in Tables A-1 and A-2 of Appendix A).

1. *Weather Conditions*: Solar radiation values are those for a clear atmosphere on August 1 at 40 deg north latitude (See Table 4, Chapter 13, THE GUIDE, 1955). The outdoor dry-bulb temperature cycle has a 95 F maximum (see Table 8 Chapter 13, THE GUIDE, 1955). Hourly temperatures are listed in Table A-1. The outdoor surface conductance for convection and radiation exchanges combined is 4.0 Btu per (hr) (sq ft) F.

2. *Exposed Wall Branch*: This is represented by the circuit w-h with the surface h in convection exchange with the room air *a*, and in radiation exchange with room surfaces *p*, *c*, and *f*. The cyclic sol-air temperature for the weather side of the exposed wall is represented by the generator symbol designated T_w. The heat capacity of the wall is concentrated in 2 lumps of equal capacitance, C_k *and* C_m.

3. *Glass-Shade Branch*: This is represented by the circuits n-g and g-b with a cyclic sol-air temperature, T_n, for the window glass, and a cyclic thermal current, I_b, for the solar radiation absorbed by the shade. These shade inputs were computed by methods outlined in Reference 6 and the values are given in Table A-1. The heat-storage capacity of glass and shade are considered negligible for the purposes of this problem. It is further postulated that the window and shade exchange heat by convection with the room air independently of each other (experience seems to indicate that this is true), that they are in radiant-energy-exchange relationship with each other, but that only the shade exchanges radiant energy with the room surfaces. They are treated as *black* surfaces in these exchanges.

4. *Floor-Ceiling Branches*: The ceiling is represented by the heat sink, Q_c, which in this problem, is assumed to be at a constant temperature of 75 F throughout the 24-hr cycle. The floor circuit *f-c'* ends with the heat sink, Q_c', of the ceiling of the room below. The floor heat capacity is contained in capacitances C_d and C_e.

5. *Partition Branch*: All interior partitions have been treated as one surface at temperature T_p. The resistance branch *p-q* contains half the thermal resistance of the partition with the corresponding thermal capacitance concentrated at the center, *s*. Point *q* is at the midpoint of the partition and is treated as an adiabatic boundary; that is, there is no heat flow between *s* and *q*.

6. *Low-Temperature Radiation Exchanges*: Shape factors and circuit resistances are those used in the illustrative problem of Reference 1 and are listed in Table A-2.

7. *Convection Exchanges*: Values are listed in Table A-2.

8. *Transmitted-Solar-Radiation Thermal Current*: This cyclic energy source is designated *I* in Fig. 1 and was computed for the glass-shade combination. This energy is postulated to leave the shade as if it were a perfectly diffusing source and to be uniformly distributed over the surfaces on which it falls. The amount absorbed by each room surface was determined in accordance with the shape factors of Table A-2 and surface reflectances given in Table 1. Allowance was made for interreflections. The total cyclic quantity of transmitted radiation is given in Table A-1 and the fraction absorbed by each surface by Table A-3.

9. *Artificial-Lighting Thermal Current*: A semi-direct fluorescent lighting system is installed in the office to provide 50 ft-candles at the working level. Data from Reference 7 were used to compute the energy distribution from the lamps and fixtures. This energy amounts to 3.6 watts per sq ft of floor area. The radiation thermal currents, both the long-wave and the short-wave visible, originate in the circuit from source, L. A convection current is also included and is connected directly to point *a*. The distribution is given in Table A-3.

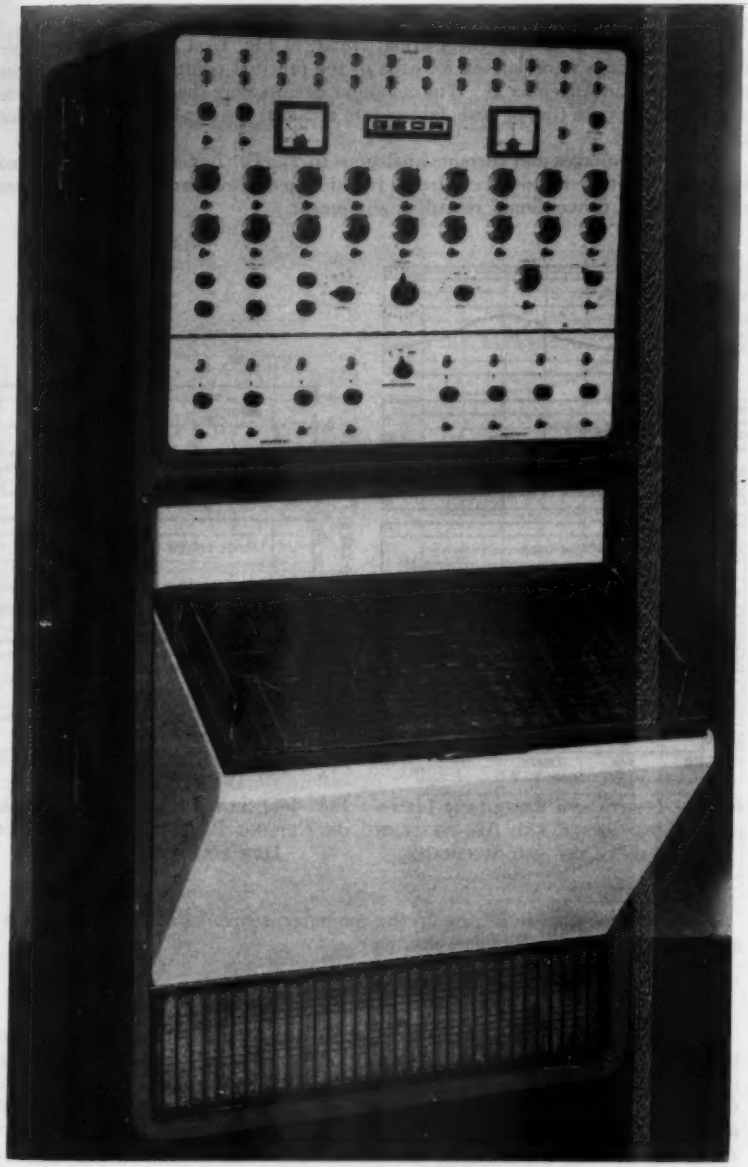

FIG. 2—THE ANALOGUE COMPUTER

Analogue Computer: The computer used in this study is illustrated in Fig. 2.

As a first step in solving the thermal circuit of Fig. 1 on the computer, heat-balance equations were derived for each of the junction points. These equations may be found in Appendix D. The computer circuits were then derived to provide a dynamic response (taking scale factors into account) identical to that of the given thermal circuit.

The input voltages and currents analogous to T_n, T_w, I_b and I were introduced into the circuit by means of electronic function generators, and the outputs were recorded with conventional recording equipment.

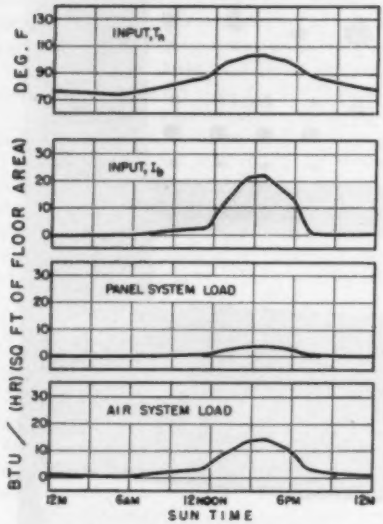

FIG. 3—INPUTS AND RESULTANT LOADS ON CEILING PANEL AND AIR SYSTEMS DUE TO GLASS AND SUNSHADE

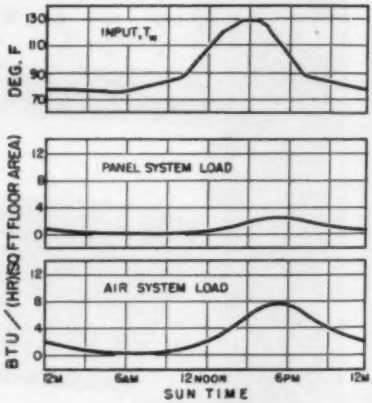

FIG. 4—INPUT AND RESULTANT LOADS ON CEILING PANEL AND AIR SYSTEMS DUE TO EXPOSED WALL

The technique used in setting up the computer is covered in the book by Korn and Korn listed in the Bibliography.

COMPUTER RESULTS

Solutions in terms of time variable circuit temperatures and instantaneous cooling loads for the thermal circuit shown in Fig. 1 were obtained by means of a standard commercial electronic differential analyzer. These were recorded on a standard 6-channel high-frequency-response recording galvanometer and are reproduced as Figs. 3, 4, 5, 6, 7 and 8. Input cycles were repeated until the output curves reached the steady periodic state. All heat-flow rates are in Btu per (hour) (square foot of floor area). The inputs T_w, I, and T_n with I_b were first generated individ-

ually so that loads due to each input could be determined separately. For example, with T_w as the input, thermal current generators I and I_b were disconnected and circuit point, n, was connected to ground around the thermal potential generator, T_n. This procedure of treating a linear system, such as the circuit is postulated to be, gives the system temperatures relative to the chosen constant reference tem-

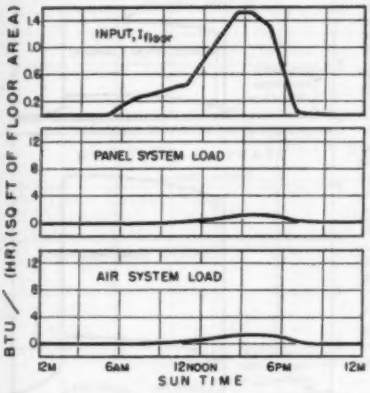

FIG. 5—SOLAR RADIATION TRANSMITTED THROUGH GLASS AND SUNSHADE AND RESULTANT LOADS ON PANEL AND AIR SYSTEMS

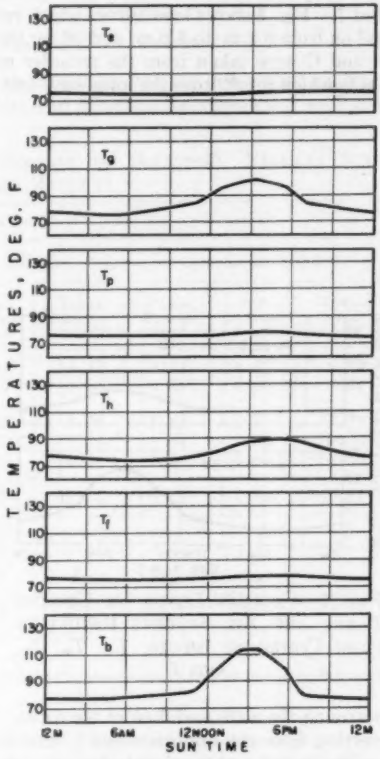

FIG. 6—TEMPERATURES IN THERMAL CIRCUIT RESULTING FROM COMBINED INPUTS T_n, T_w, I_b, AND I

perature. (See Reference 8 for proof.) The resulting cooling loads are the algebraic sums of the thermal currents to point a, room air temperature, and points c and c', ceiling panel temperature.

Fig. 3 gives the loads on the air system, Q_a, and the ceiling system, $(Q_c + Q_{c'})$, due only to the sol-air temperature, T_n, for the glass and the solar radiation absorbed by the shade, I_b. It also shows the input curves which were generated elec-

tronically as a series of straight-line segments. Comparison of the curves with the values in Table A-1 shows good agreement. Fig. 4 shows the resultant loads due to the sol-air temperature for the exposed wall and also the input curve, T_w. Fig. 5 shows the load components which result from the solar radiation transmitted through the window shade. Only the input curve for the floor is shown, as the others are proportional. Finally, these input functions were generated simultaneously and the resulting temperatures and heat-flow rates are shown in Figs. 6 and 7. Fig. 8 shows load curves which result from the lighting system. This was full on from 8 a.m. to 8 p.m. and off for the remainder of the 24-hr period. Curves A and C were taken from the recorder records and are the panel and air-system loads which result from the long-wave and short-wave radiation from the luminaire

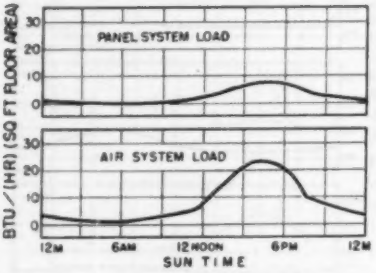

FIG. 7—COOLING LOADS ON CEILING PANEL AND AIR SYSTEMS RESULTING FROM COMBINED INPUTS, T_n, T_w, I_b, AND I

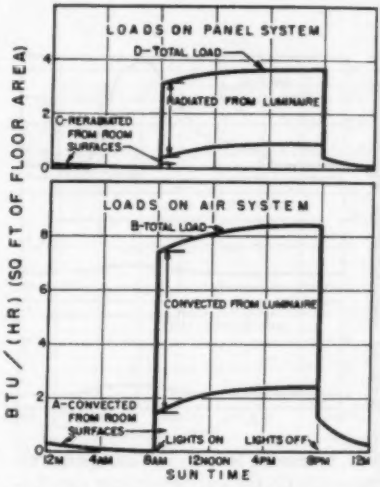

FIG. 8—COOLING LOADS ON PANEL AND AIR SYSTEMS RESULTING FROM ARTIFICIAL LIGHTING

warming the walls and floor of the room. Curve B includes the direct load by convection from the luminaire and Curve D includes all radiation which reaches the ceiling panel and is absorbed there, whether it arrives there directly from the luminaire or by reflection from the room surfaces.

EVALUATION OF DIFFERENTIAL ANALYZER ACCURACY

To provide a basis for judging the accuracy of the analyzer results, the circuit diagram was solved for steady-state inputs using the 24-hr average values given in Table A-1 and the thermal currents in Table A-3. This calculation was simplified by the fact that the net heat storage over a 24-hr cycle is zero, and thermal capacitance could therefore be disregarded.

The results of the steady-state computations, together with the 24-hr average values of heat-flow rates and surface temperatures determined from the analyzer

curves, are shown in Table 2. It will be noted that agreement between the 2 sets of heat-flow values is excellent except for the load component due to transmitted solar radiation. Inspection of the curves shows that a small zero error in recording could easily account for the error. This could have been prevented by a better choice of scale for this pair of output curves. With inputs combined, 24-hr averages determined by computation and from averages of the analyzer curves, differ by less than 5 percent. Average surface temperatures also compare favorably.

DISCUSSION

The results which have been obtained demonstrate that the differential analyzer can be applied to solving practical problems of considerable complexity and give

TABLE 2—COMPARISON OF 24-HR AVERAGES OF ANALYZER RESULTS WITH CALCULATED RESULTS

Comparison of Heat Flow Rates Btu per (Hour)(square foot of floor area)

COMPONENT	ANALYZER RESULTS			CALCULATED RESULTS		
	PANEL SYSTEM LOAD	AIR SYSTEM LOAD	TOTAL LOAD	PANEL SYSTEM LOAD	AIR SYSTEM LOAD	TOTAL LOAD
A Glass & Shade (Fig. 3)	1.07	4.45	5.52	1.18	4.40	5.58
B Transmitted Solar (Fig. 5)	0.29	0.37	0.66	0.39	0.50	0.89
C Exposed Wall (Fig. 4)	0.90	2.88	3.78	1.12	2.77	3.89
A + B + C	2.26	7.70	9.96	2.64	7.66	10.30
A, B + C Combined (Fig. 7)	2.58	7.95	10.53	2.64	7.66	10.30
Lights[a] (Fig. 8)	0.47	1.27	1.74	0.42	1.10	1.52

Comparison of Surface Temperatures (Degrees above base temperature at 75 F)

	T_b	T_g	T_h	T_p	T_f	
Analyzer	11.7	9.2	5.1	1.7	1.0	
Calculated	11.5	9.5	5.8	1.4	0.8	

[a] Loads due to radiation from luminaires which is absorbed by room surfaces but not including direct-to-ceiling load.

engineering results with satisfactory precision. One should not lose sight of the fact, however, that the problem was set up by applying thermal-circuit techniques and that the analyzer was simply a tool. It should be noted that other operations besides those demonstrated can be carried out with the aid of an analyzer. For example, a constant load could have been imposed on the air system and the resultant surface and room air temperature determined. This load could be cycled so that the room air temperature could be held between predetermined upper and lower limits (see Reference 5). Regimes that could be studied are schedules of air-system loads and of ceiling-panel temperatures. With little additional time, all of the circuit parameters could be varied over the desired ranges, as long as the circuit remains the same. This is perhaps one of the most important features in the use of the analyzer (or a direct analogue). Changes in the circuit form would

require alterations in the connections of circuit resistances, capacitances, inputs and operational amplifiers.

One limitation evident from the experience gained with the analyzer is that a circuit of greater complexity, for example, one with more capacitance lumps in the exposed wall branch or with the partition walls subdivided, would require the use of more operational amplifiers. In this particular example, 20 of the machine's complement of 24 were used.

Whether or not the thermal capacitance lumping used in this illustrative problems is adequate cannot be answered exactly (see Reference 9). As a check, the periodic heat flow through the exposed wall was computed for the lumping shown in Fig. 1 for constant radiation and convection heat sinks. The exact solution

TABLE 3—SUMMARY OF PEAK SYSTEM LOADS, TIMES OF PEAKS AND COMPARISONS WITH INSTANTANEOUS GAIN

Heat Flows in Btu Per (hr)(sq ft of Floor)

COMPONENT	CEILING SYSTEM		AIR SYSTEM		COMBINED SYSTEM		CALCULATED INSTANTANEOUS GAIN[a]		LOAD GAIN
	MAX LOAD	TIME OF MAX	MAX LOAD	TIME OF MAX	MAX LOAD	TIME OF MAX	MAX GAIN	TIME OF MAX	
Glass & Shade, A Transmitted	3.5	3 pm	14.5	3:30 pm	18.0	3:30 pm	21.3	3 pm	85
Solar, B	1.0	3:30 pm	1.4	3:30 pm	2.4	3:30 pm	3.4	3 pm	70
Exposed Wall, C	2.6	5 pm	7.8	5 pm	10.4	5 pm	12.2	5 pm	85
A, B & C Combined	7.2	4:30 pm	23.0	4:15 pm	30.2	4:15 pm			
Lighting: Start	3.1		7.4		10.5		12.3		85
Equilibrium	3.7	6 pm[b]	8.4	6 pm[b]	12.1	6 pm[b]	12.3		98

[a] Method used in THE GUIDE.
[b] Approximate time of equilibrium.

was then computed by H. A. Johnson's[10] charts, and it was found that the results were substantially the same.

PRACTICAL RESULTS

Maximum Cooling Loads: Table 3 summarizes the results in terms of maximum loads on the 2 systems and times of peak loads. These data are needed in sizing the components of the 2 cooling circuits. The calculated instantaneous rates of heat gain (by method in THE GUIDE) are also listed in Table 3 for comparison. As solutions of other circuit problems have shown, the ratios of maximum cooling loads to the maximum rates of instantaneous gain for periodically varying inputs are substantially less than unity. In this case they are 0.70 for the transmitted solar radiation and 0.85 for the other inputs. This is due largely to the damping effect of the internal heat storage capacity. It should be pointed out that the choice of this relatively light structure does not bring out the differences between instantaneous gain and cooling load that would be expected with heavy masonry construction. Times of peaks are about one-half hour later than calculated, which is not a significant difference.

The load due to the steadily applied lighting, follows a somewhat different pattern. At the start of the *lights on* period, the load is 85 percent of the instantaneous

energy input to the luminaries, but at 6 p.m., and near equilibrium, it has become almost 100 percent. Fig. 8 shows that the load drops rapidly at *lights off* but does not become zero until nearly time for the lights to go on again.

Ceiling Panel Performance: Of practical interest is the performance of the ceiling panel in picking up radiant energy loads. The analyzer results for the lights and transmitted solar radiation at the time of maximum total load are given in Table 4. Calculated results for the 24-hr period are also included. Discussion will be confined to the calculated results, because the analyzer results for the transmitted solar radiation seem to include some zero error. Total radiant pickup was 42 percent of the total transmitted solar load and 57 percent of the total radiant output of the lights. In terms of the total energy input to the lighting system (1.8 watts

TABLE 4—COMPONENTS OF RADIANT ENERGY LOAD ON CEILING PANEL AT 75 F

Loads are in Btu/(hr)(sq ft of floor)

TRANSMITTED SOLAR RADIATION	AT 3 P.M.[a]		24-HR AVERAGE[b]	
	LOAD	%	LOAD	%
Total Transmitted	3.42	100	0.93	100
Direct to panel	0.56	16	0.16	17
Re-radiation to panel	0.44[c]	13	0.23	25
Total to panel	1.00	29	0.39	42
Radiation from Luminaires				
Total Radiant Output	6.30	100	3.15	100
Direct to panel	2.75	44	1.38	44
Re-radiation to panel	0.90	14	0.42	13
Total to panel	3.65	58	1.80	57

[a] Analyzer results. [b] Calculated results. [c] Chart record value—0.56.

per sq ft average for the 24-hr period,) the total radiant pickup was 29 percent. These results are dependent upon the assumed distribution of the transmitted solar radiation and the input to the luminaires. The distribution of the transmitted solar energy is believed to closely approximate what would be found in practice. The distribution of the energy input to the luminaire into long-wave and short-wave radiation and convection is admittedly an estimate, but a rational one. This distribution is given in part B of Table A-3.

Note that the results given in Table 4 are for a panel temperature of 75 F, which is the same as the room air temperature; hence there is no convection gain by the panel. In actual practice, however, an exchange will take place between the panel and the supply air, which must be introduced at a temperature below that of the room air. This exchange will be influenced by the manner in which the air is introduced. This is a complex problem beyond the scope of the illustrative problem. However, research on this point and also on the matter of energy distribution from luminaires has been carried on at the ASHAE Research Laboratory as part of the panel heating and cooling program and will be reported soon.

The effect of operating the panel at a different temperature can be determined from the steady-state solutions of the circuit equations. (See Appendix C.) If it be assumed that convection conductances for panel and room surfaces remain

unchanged, each degree decrease in panel temperature will increase the panel load and decrease the air-system load by constant amounts. Room-surface temperatures will also change by constant amounts. These changes are summarized in Table 5. Note that there is a small increase in the total load. This results from the fact that the temperature of the radiant environment seen by the exposed wall and window is lower. Note also that a one-degree reduction in panel temperature would cause the total air-system load to become negative during the night if both systems were kept in continuous operation. This would require warm rather than cool air to be supplied to the space.

ACKNOWLEDGMENT

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TABLE 5—CHANGES IN LOAD DISTRIBUTION AND ROOM SURFACE TEMPERATURES CAUSED BY A DECREASE OF ONE DEGREE IN CEILING PANEL TEMPERATURE

REDUCTION IN ROOM-SURFACE TEMPERATURES, DEGREES F			
Floor	0.33	Shade	0.18
Partitions	0.27	Glass	0.03
Exposed Wall	0.19		

CHANGES IN LOAD DISTRIBUTION, BTU/(HR) (SQ FT FLOOR AREA)	
Reduction in air system load due to reduction of room surface temperatures	0.71
Reduction in air system load due to heat convected to panel	1.10
total reduction	1.81

INCREASE IN PANEL SYSTEM LOAD, BTU/(HR) (SQ FT)	
Increase in panel system load due to net increase of radiation exchange	0.80
Increase in panel system load due to heat convected to panel	1.10
total increase	1.90

INCREASE IN TOTAL COOLING LOAD 1.90 − 1.81 = 0.09 BTU/(HR) (SQ FT)

CONCLUSIONS

1. The electronic differential analyzer with its associated input generators and recorders was found to be capable of solving a complex thermal-circuit problem with an accuracy entirely suitable for making detailed analyses of the behavior of air-conditioning thermal systems, expressed in the form of thermal-circuit diagrams. The combination of the thermal-circuit technique and the computational ability of a differential analyzer showed itself to be a powerful tool in solving the complex air-conditioning thermal problems. It is regrettable that the electrical circuit analogue is not as systematically applied to heat-transfer problems as it has been applied, in the last decades, to mechanics, hydraulics and acoustics. In heat-transfer investigations, partial differential equations are always involved, and with usually irregular boundary conditions, these problems are mostly unsolvable. Thus the thermal circuit technique with the aid of a differential analyzer could be an invaluable help.

2. The analyzer is suitable for handling systems of greater complexity, but additional operational amplifiers would be needed. It is evident that cost must place a practical

limit to the number which could be used. Limitations are closely related to the fineness of the capacitance lumping required. Though there has been as yet no rational way of determining the lumping requirements for any given problem, the question of the necessary minimum number of lumps and of their most effective distribution along a circuit branch was recently investigated (see References 9 and 11) and substantial progress was made.

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APPENDIX A

TABLE A-1—SELECTED WEATHER CONDITIONS AND RESULTANT THERMAL CIRCUIT INPUTS

SUN TIME	WEATHER CONDITIONS			THERMAL CIRCUIT INPUTS			
	DRY BULB, F	INCIDENT SOLAR RADIATION, BTU/(HR.)(SQ FT WINDOW)		SOL-AIR TEMPS, F		SOLAR RADIATION THERMAL CURRENTS, BTU/(HR.)(SQ FT FLOOR)	
		DIRECT	DIFFUSE	WALL, T_w	WINDOW, T_s	TRANS. BY WINDOW, I	ABSORBED BY SHADE, I_b
12 mid	77			77	77		
1 am	76			76	76		
2	76			76	76		
3	75			75	75		
4	74			74	74		
5	74		1	74	74	0.0	0.0
6	74		8	75	74	0.3	0.6
7	75		13	77	76	0.5	1.1
8	77		16	80	78	0.7	1.3
9	80		18	83	81	0.8	1.5
10	83		20	87	84	0.8	1.7
11	87	22	24	95	87	1.0	2.7
12 noon	90	76	26	106	94	1.6	8.0
1 pm	93	121	29	119	100	2.4	13.7
2	94	151	32	126	102	3.0	18.4
3	95	164	33	129	104	3.4	21.1
4	94	156	31	127	102	3.4	20.3
5	93	125	26	119	100	2.9	16.7
6	91	69	15	106	95	1.8	12.1
7	87	3	1	88	87	0.1	0.3
8	85			85	85		
9	83			83	83		
10	81			81	81		
11	79			79	79		
24-Hr Avg.	83.1			91.6	85.3	0.9	5.0

TABLE A-2—THERMAL CIRCUIT CONSTANTS

RESISTANCES, R, IN DEG F PER BTU HR				CAPACITANCES, C, IN BTU PER DEG F	
CONVECTION	RADIATION	CONDUCTION			
a-c 0.91	c-f 2.57	k-m 1.39		C_s	0.42
a-f 0.91	c-b 15.10	m-h 0.70		C_d	2.62
a-b 4.70[b]	c-h 8.43	f-d 0.38		C_e	2.77
a-h 3.18	c-p 2.85	d-e 0.76		C_k	1.58
a-p 1.09	f-b 17.20	e-c[1] 3.79		C_m	1.58
a-g 9.23	f-h 7.87	p-s 2.23			
g-n 1.15	f-p 2.85	w-k 0.70[a]			
	b-p 13.10				
	b-g 4.97				
	p-h 5.81				

$R_{convection}$ = (floor area)/(surface area)(convection conductance)
$R_{conduction}$ = (floor area)/(surface area)(k/1)
$R_{radiation}$ = (1.08)(floor area)/(surface area)(shape factor)
$C_{capacitance}$ = (surface area)(1)(ρ C_p)/(floor area)

SHAPE FACTORS, F[c]			CONVECTION CONDUCTANCES, IN BTU/(HR)(SQ FT) (F)	
F_{e-f} 0.42	F_{f-h} 0.14		Outdoor	4.0
F_{e-b} 0.07	F_{f-p} 0.38		Walls	0.7
F_{e-h} 0.13	F_{b-p} 0.38		Floor	1.0
F_{e-p} 0.38	F_{b-g} 1.00		Ceiling	1.0
F_{f-b} 0.06	F_{p-h} 0.18		Glass	0.5
			Shade	0.4

[a] Includes outdoor surface resistance.
[b] Shade surface area—2.4 times window area.
[c] $A_p f_{p-h} = A_h F_{h-p}$, etc.

TABLE A-3—DISTRIBUTION OF THERMAL CURRENTS

A. DISTRIBUTION OF TRANSMITTED SOLAR RADIATION

SURFACE	FRACTION ABSORBED
Ceiling, c	0.165
Partitions, p	0.316
Floor, f	0.459
Exposed Wall, h	0.037
Window Shade, b	0.023

B. DISTRIBUTION OF TOTAL ENERGY SUPPLIED TO LIGHTING SYSTEM

BTU PER (HR)(SQ FT FLOOR AREA)

CIRCUIT POINT	RADIATION			CONVECTION
	SHORT-WAVE	LONG-WAVE	TOTAL	
Ceiling, c	0.19	2.56	2.75	
Partitions, p	0.24	0.98	1.22	
Floor, f	0.61	1.08	1.69	
Exposed Wall, h	0.08	0.33	0.41	
Window Shade, b	0.05	0.18	0.23	
Room Air, a				5.98
Totals	1.17	5.13	6.30	5.98
Percent of radiation plus convection	9.5	41.8	51.3	48.7

APPENDIX B

SOLVING THERMAL CIRCUITS WITH AN ELECTRONIC DIFFERENTIAL ANALYZER

The thermal-circuit diagram of Fig. 1 emphasizes the complex inter-relationship which exists between the temperatures of all the elements of the thermal system under study. Quantitatively, these temperatures depend upon the characteristics of the heat sources (sun, outdoor air, lights), and heat sinks (cooling system), and on the values of circuit parameters (thermal resistances and capacitances). Unknown temperatures could be found by solving the corresponding system of nodal equations. When the temperatures are known, the needed heat-flow rates can be computed. Although the circuit equations can be written easily enough, analytical solution is often too cumbersome if more than a few equations are to be solved. The thermal circuit of Fig. 1 requires, for example, a system of 10 equations (five algebraical, and five differential). For this and similar complicated circuits, solution by means of special analogue or computing equipment is advantageous, and many times unavoidable.

The differential analyzer is an analogue computer which can perform the limiting processes; hence it is capable of differentiation, integration, and of the solution of certain groups of differential and integral equations. In the differential analyzer, voltage is analogous to the dependent variable; and differentiation and integration are made with respect to *machine time* as the independent variable.

Consider the circuit branch *w-h* of Fig. 1 with *h* connected to a heat sink at constant temperature T_v. The circuit diagram for this branch is shown in Fig. B-1a. It is required to find the variation of temperatures T_k and T_m with time when thermal resistances R_{wk}, R_{km}, R_{mv} and thermal capacitances C_k, and C_m are given.

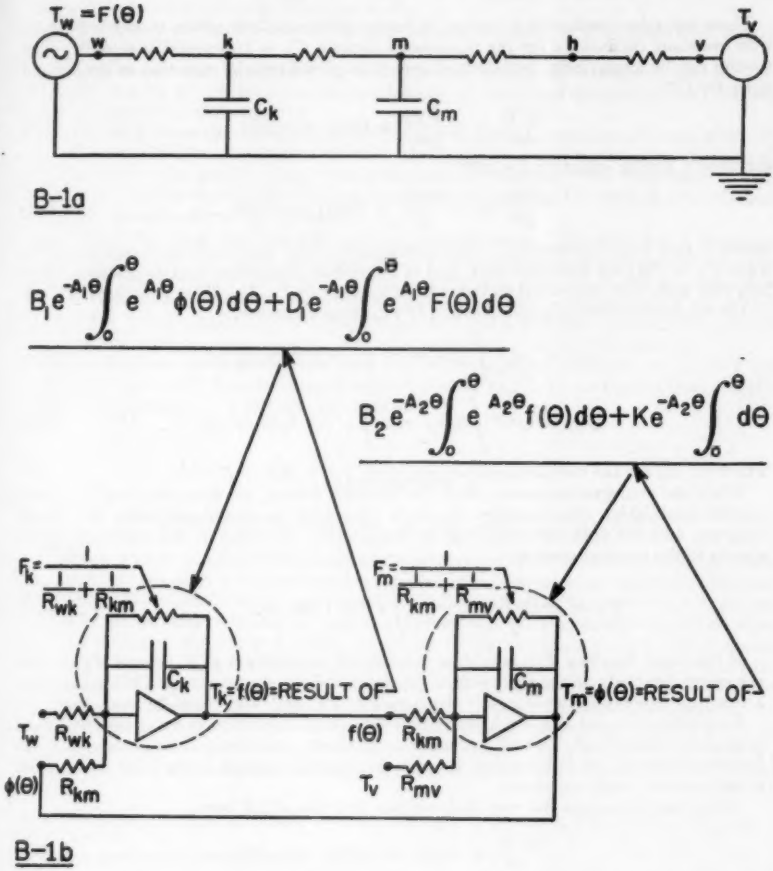

FIG. B-1—A SIMPLE THERMAL CIRCUIT AND THE CORRESPONDING DIFFERENTIAL ANALYZER CIRCUIT

Using Kirchhoff's first law for the circuit Fig. 3, 2 nodal equations can be written

$$\frac{T_w - T_k}{R_{wk}} = C_k \frac{dT_k}{d\theta} + \frac{T_k - T_m}{R_{km}}; \quad \ldots\ldots\ldots \quad \text{(B-1)}$$

$$\frac{T_k - T_m}{R_{km}} = C_m \frac{dT_m}{d\theta} + \frac{T_m - T_v}{R_{mv}}; \quad \ldots\ldots\ldots \quad \text{(B-2)}$$

These equations represent a system of linear differential equations of the first order with constant coefficients for the unknown functions $T_k = f(\theta)$ and $T_m = \phi(\theta)$. The system can be analytically transformed into 1 linear differential equation of the second order for $f(\theta)$:

$$\frac{d^2f(\theta)}{d\theta^2} + A \frac{df(\theta)}{d\theta} + Bf(\theta) = F_k(\theta); \quad \ldots\ldots\ldots \quad \text{(B-3)}$$

and into a similar equation for $\phi(\theta)$

$$\frac{d^2\phi(\theta)}{d\theta^2} + A \frac{d\phi(\theta)}{d\theta} + B\phi(\theta) = F_m(\theta); \quad \ldots\ldots\ldots \quad \text{(B-4)}$$

where A and B are functions of circuit parameters; $F_k(\theta)$ and $F_m(\theta)$ are functions of the input $T_w = F(\theta)$, its first derivative, and of the circuit parameters and constants. Symbols $f(\theta)$, $\phi(\theta)$, $F(\theta)$ were used to stress the character of T_k, T_m, T_w as functions of time.

The solution of the Equations B-3 and B-4 in integral form is:

$$f(\theta) = e^{m_2\theta} \int [e^{(m_1-m_2)\theta} \int e^{-m_1\theta} F_k(\theta) d\theta] \, d\theta; \quad \ldots\ldots\ldots \quad \text{(B-5)}$$

$$\phi(\theta) = e^{m_2\theta} \int [e^{(m_1-m_2)\theta} \int e^{-m_1\theta} F_m(\theta) d\theta] \, d\theta; \quad \ldots\ldots\ldots \quad \text{(B-6)}$$

where m_1, m_2 are the roots of auxiliary equation: $m^2 + Am + B = 0$.

When the analogue computer circuit is correctly drawn, machine values of the parameters established, the necessary elements connected in accordance with the circuit diagram, and the potentiometers rightly fixed,—then the *setup* of the computer corresponds to the integral symbol:

$$[\,] = e^{m_2\theta} \int [e^{(m_1-m_2)\theta} \int e^{-m_1\theta} [\,] d\theta] \, d\theta. \quad \ldots\ldots\ldots \quad \text{(B-7)}$$

If the input function $F(\theta)$ would be introduced [in the form of $F_k(\theta)$ and $F_m(\theta)$] into the empty brackets on the right side of Equation B-7, or, in the case of a computer, into a function generator, the integrals the functions $f(\theta)$ and $\phi(\theta)$ could be evaluated.

Equations B-3 and B-4, each containing one unknown function only, can be solved separately (analytically as well as by the computer). But the real advantage of a differential analyzer lies in its ability to solve directly the original system, i.e. to integrate simultaneously both equations.

When the Equations B-1 and B-2, written in a simplified form:

$$\frac{df(\theta)}{d\theta} + A_1 f(\theta) = B_1\phi(\theta) + D_1F(\theta); \quad \ldots\ldots\ldots \quad \text{(B-8)}$$

$$\frac{d\phi(\theta)}{d\theta} + A_2\phi(\theta) = B_2 f(\theta) + K; \quad \ldots\ldots\ldots \quad \text{(B-9)}$$

are solved separately as if $f(\theta)$ and $\phi(\theta)$ on the right side of the equations were known functions, the result will be:

$$f(\theta) = B_1 e^{-A_1\theta} \int_0^\theta e^{A_1\theta} \phi(\theta) \, d\theta + D_1 e^{-A_1\theta} \int_0^\theta e^{A_1\theta} F(\theta) \, d\theta; \quad \ldots\ldots\ldots \quad \text{(B-10)}$$

$$\phi(\theta) = B_2 e^{-A_2\theta} \int_0^\theta e^{A_2\theta} f(\theta)\, d\theta + K e^{-A_2\theta} \int_0^\theta d\theta \quad \text{(B-11)}$$

This is a system of 2 linear integral equations of the third kind (Volterra's type) for unknown functions $f(\theta)$ and $\phi(\theta)$, where A_1, B_1, D_1 and A_2, B_2 are functions of circuit parameters (R_{wk}, R_{km}, R_{mv}, C_k, C_m); the value of K depends upon the parameters and upon the choice of constant temperature T_v.

The diagram of Fig. B-1b shows in rough outline the function of a differential analyzer in the present example.

The real processes in computing elements can hardly be visualized, and no exact parallel can be drawn between the mechanism of analytical integration and the integrating process in the computer.

Analytical integration consists of relatively few logico-mathematical steps, every one of which can be represented by a mathematical symbol. The last step, mathematically symbolized, is the result. Analytical integration is timeless. The time spent for consideration and for writing the symbols has obviously nothing in common with the independent variable, θ, of the problem.

In the differential analyzer there are no separate steps. The function is integrated continuously, point by point, and the result is symbolized on a strip of coordinate paper. The time necessary for integration is the independent variable of the given problem. Step-by-step preparation of the correct *setup* of an analogue computer is not the work of a machine, but of an engineer or a mathematician, and it is in fact a particular analytical method dictated by the principles and construction of a differential analyzer.

To understand the *mathematical activity* of the computing elements in Fig. B-1, one may simply imagine that the integral equations B-10 and B-11 are there solved in a self-balancing manner by a continuous iteration.

APPENDIX C

STEADY-STATE SOLUTION OF THE CIRCUIT EQUATIONS

Surface Temperatures:—The steady-state solution of the circuit equations given in Appendix D provides a means of extending, exactly in some cases and approximately in others, the results obtained with the differential analyzer. The solution is expressed in terms of coefficients, which are listed in Table C-1. The coefficient times the 24-hr average value of the input function gives the temperature component which results from that input. The coefficients for thermal-current inputs Q_t, Q_p, and so on, have units deg F/[(Btu)(hr sq ft floor)]. Those for the thermal potential inputs, T_n and T_w, are dimensionless. An example will illustrate the use of these coefficients.

The 24-hr average of T_n and T_w are 85.3 F and 91.6 F respectively and for I and I_b are 0.9 and 5.0 Btu per (hr)(sq ft of floor) respectively. (See Table A-1). With T_a and T_c at 75 F, the temperature of point p, the partition wall surface, due to T_n and T_w is:

$$T_p = [0.012 \times 85.3] + [0.036 \times 91.6] + [0.678 \times 75] + [0.273 \times 75] = 75.72 \text{ F}$$

The temperature increment due to I and I_b is:

$$\Delta T_p = 0.9[0.316 \times 0.567 + 0.037 \times 0.101 + 0.459 \times 0.098] + [0.084 \times 5.0] = 0.63$$

Note that coefficients 0.316, 0.037, and 0.459 in the first term are distribution factors from Table A-3 for the transmitted solar radiation. The value 5.0 in the second term includes transmitted radiation reflected back to the window shade and absorbed there. The total temperature is 1.4 deg above the reference temperature 75 F and is the value listed for T_p in Table 2 under *Comparison of Surface Temperatures*. A useful check is that the sum of the coefficients of the potential inputs (T_n, T_w, T_a, and T_c) equals unity.

If the ceiling panel temperature were maintained steadily at 65 F throughout 24 hours instead of at 75 F, T_p would be (75–60) 0.325 or 3.25 deg lower.

Heat-Flow Rates: Heat flow from the partition walls to the ceiling panel and to the air are respectively:

$$Q_c = (76.4 - 75)/2.85 = 0.49 \text{ Btu/(hr)(sq ft floor)}$$

$$Q_a = (76.4 - 75)/1.09 = 1.28 \text{ Btu/(hr)(sq ft floor)}$$

where 2.85 and 1.09 are radiation and convection exchange resistances respectively from Table A-2.

It is seen that temperatures and cooling loads on a 24-hr basis could be computed by means of the data in Tables C-1 and A-2 for quite different inputs and distributions. The periodic heat-flow rates would differ, however, unless the new input curves were similar in shape to those of the illustrative problem.

TABLE C-1—STEADY-STATE COEFFICIENTS OF INPUT FUNCTIONS FOR ROOM SURFACE TEMPERATURE

SURFACE	INPUT							
	Q_p*	Q_h*	Q_f*	Q_b*	T_a	T_w	T_b	T_s
p	0.567	0.101	0.098	0.084	0.012	0.036	0.678	0.273
h	0.100	0.941	0.069	0.020	0.003	0.333	0.471	0.190
f	0.097	0.070	0.470	0.059	0.009	0.025	0.643	0.325
b	0.084	0.020	0.060	1.737	0.258	0.008	0.550	0.183
g	0.014	0.003	0.010	0.297	0.781	0.001	0.186	0.031

* Q may be a transmitted solar radiation input or radiant energy from lights, or any other radiant input

APPENDIX D

COMPARISON OF THREE DIFFERENT METHODS OF THE SOLUTION OF THERMAL PROBLEMS

Consider a simple homogeneous wall (Fig. D-1a) that is uniformly heated by a periodic heat source on the side A; the periodic variation of temperature on B is to be calculated. Initial conditions are: On side A, $T = 0$ for $\theta = 0$; $T = T_0 \cos \omega\theta$ for $\theta > 0$; the wall has a temperature $T = 0$ for $\theta = 0$; the space S (sink) has the temperature $T = K$ (constant) for $\theta \geqq 0$ (T = temperature, θ = time, ω = angular frequency). The exact solution of the problem requires the use of Fourier's heat conduction equation, in this case in one-dimensional form:

$$\frac{\partial T}{\partial \theta} = \alpha \frac{\partial^2 T}{\partial X^2}$$

where X is the distance from side A ($0 \leqq X \leqq l$), l is the wall between, α is the thermal diffusivity $\left(\alpha = \frac{K}{\rho C_p}\right)$; K = thermal conductivity, ρ = density, C_p = specific heat of the wall.

The solution, even for very simple boundary conditions, is rather complicated. (See, for example, Carslaw-Jaeger: *Conduction of Heat in Solids*, Oxford University Press, pp. 88–92).

The exact solution would be very simple, if $l \to \infty$ (for the infinitely thick wall):

$$T_m = T_o e^{-X\sqrt{(\omega/2\alpha)}} \cos\left(\omega\theta - X\sqrt{\frac{\omega}{2\alpha}}\right)$$

This result can be used with sufficient approximation for very thick walls. If the wall can be considered as infinitely thick when the amplitude at surface B is damped to 1 percent of the amplitude at surface A, the foregoing formula may be used.

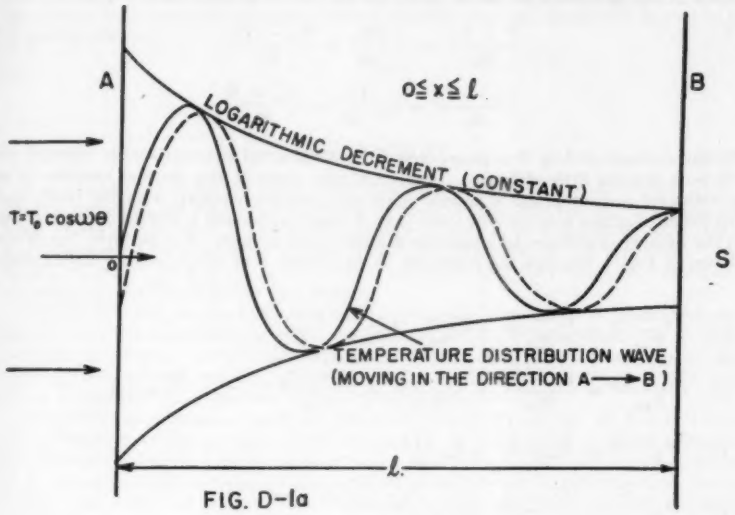

FIG. D-1a

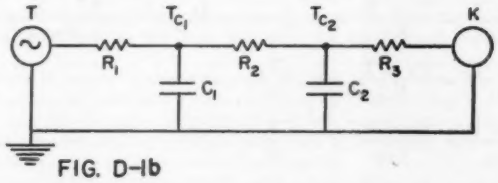

FIG. D-1b

FIG. D-1—A WALL SECTION WITH PERIODIC INPUT AND ITS CORRESPONDING THERMAL CIRCUIT

The corresponding thickness of this *infinite wall* is $1 = 2.6\sqrt{\alpha\theta_o}$ where θ_o is the period of the temperature oscillation. Actual building walls are seldom of *infinite thickness* according to this definition. (See M. Jakob: *Heat Transfer*, J. Wiley, New York, 1949, pp. 296–297).

To surmount these difficulties and to avoid partial differential equations with complicated boundary conditions, there was introduced the concept of the lumped thermal circuit. From the previous example, the thermal circuit of Fig. D-1b would be obtained by dividing the wall thickness into 2 halves and lumping the thermal capacitance of each half at the centerpoint of the half. The resistances are calculated as the thermal resistance between internal wall points and between internal wall points and surfaces.

The lumped thermal circuit is only an approximate representation of the real case, but it has been found that for a very close approximation it is necessary to use relatively few lumps. Generally, 2 or 3 lumps are quite sufficient. Such a thermal circuit as shown in the figure can be solved easily by the following equations: (Fig. D-1b).

$$\frac{T - T_{C1}}{R_1} = C_1 \frac{dT_{C1}}{d\theta} + \frac{T_{C1} - T_{C2}}{R_2};$$

$$\frac{T_{C1} - T_{C2}}{R_2} = C_2 \frac{dT_{C2}}{d\theta} + \frac{T_{C2} - K}{R_3}.$$

Similar systems of 2 or 3 ordinary first-order differential equations with constant coefficients present little difficulty. Unfortunately, most of the thermal systems to be investigated are composed of several parts that require lumping, with the result that the circuit diagram may include more than 3 capacitances and a corresponding increase in the number of differential equations describing the system. For example, the circuit shown in Fig. 1 requires the following 10 equations, 5 of which include differentials.

1.) $\frac{T_a - T_g}{R_{ag}} = \frac{T_g - T_b}{R_{gb}} + \frac{T_g - T_a}{R_{ga}};$

2.) $\frac{T_g - T_b}{R_{gb}} = \frac{T_b - T_e}{R_{be}} + \frac{T_b - T_p}{R_{bp}} + \frac{T_b - T_a}{R_{ba}} + \frac{T_b - T_f}{R_{bf}} - I_b;$

3.) $\frac{T_w - T_k}{R_{wk}} = \frac{T_k - T_m}{R_{km}} + C_k \frac{dT_k}{d\theta};$

4.) $\frac{T_k - T_m}{R_{km}} = \frac{T_m - T_h}{R_{mh}} + C_m \frac{dT_m}{d\theta};$

5.) $\frac{T_m - T_h}{R_{mh}} = \frac{T_h - T_e}{R_{he}} + \frac{T_h - T_a}{R_{ha}} + \frac{T_h - T_p}{R_{hp}} + \frac{T_h - T_f}{R_{hf}} - I;$

6.) $\frac{T_h - T_f}{R_{hf}} = \frac{T_f - T_b}{R_{fb}} + \frac{T_f - T_a}{R_{fa}} + \frac{T_f - T_e}{R_{fe}} + \frac{T_f - T_p}{R_{fp}} + \frac{T_f - T_d}{R_{fd}} - I;$

7.) $\frac{T_f - T_d}{R_{fd}} = \frac{T_d - T_e}{R_{de}} + C_d \frac{dT_d}{d\theta};$

8.) $\frac{T_d - T_e}{R_{de}} = \frac{T_e - T_e'}{R_{ee}'} + C_e \frac{dT_e}{d\theta};$

9.) $\frac{T_f - T_p}{R_{fp}} = \frac{T_p - T_e}{R_{pe}} + \frac{T_p - T_b}{R_{pb}} + \frac{T_p - T_a}{R_{pa}} + \frac{T_p - T_h}{R_{ph}} + \frac{T_p - T_s}{R_{ps}} - I;$

10.) $\frac{T_p - T_s}{R_{ps}} = C_s \frac{dT_s}{d\theta};$

To point out the difficulty of solving these equations, it will be noted that in any real thermal system given numerically, the resistances and capacitances are 3 or 4 figure

numbers, and the inputs T_n, T_w, I and I_b are expressed by 3, 4, 5, or more terms of a Fourier series.

To prepare the equations for differential analyzer solution obviously requires careful calculation of the machine values of such factors as resistance, capacitance, feedback resistance and time scale; but whereas this work is directly proportional to the number of equations of the system, the work spent on the standard solution is proportional to the square of the number of equations: calculative work for analogue computer is only n-th part of the work with standard solution, if n is the number of equations. Because the standard analogue computers are adapted to the solution of 20–25 simultaneous equations, the immense advantages of the differential analyzer should not be overlooked or underestimated.

DISCUSSION

H. T. GILKEY, Cleveland, Ohio (WRITTEN): This paper represents another step in the exploration by the Society of the thermal circuit technique of relating heat gains and losses to equipment loads. The Society, and particularly the Laboratory Staff, are to be congratulated for exhibiting continued interest in the use of this technique. This interest is shared by a great many Society members, and they too have contributed materially to the literature on this subject.

Most of the work on this technique has been done in laboratories or research departments. Care has been taken to point out that the simultaneous equations of analogue methods can often be solved using a desk calculator. This paper shows that an electronic differential analyzer can be used. In other studies the use of punch card techniques in solving the equations has been suggested.

The authors of each paper have also been careful to point out that there is still much work to be done on the use of analogue techniques. It is necessary that this subject be investigated further so that the proper input, resistance, and capacitance values can be determined for specific cases. An incomplete tool is only little better than having no tool at all, and, if we are to realize the benefits which can accrue from the work which has already been done, it must be carried much further than it has now been taken. We can not suggest that it must be carried to the point of complete knowledge. I doubt that our knowledge of this or any subject is now or ever will be complete.

Having a tool is not enough. The tool must be so constructed that it is usable. It must be so designed that, with reasonable training, a mere human being can use it. It must be so manufactured that its use brings about economic advantage to its owner. In general, these criteria apply to the tools of engineering just as they apply to the tools of production.

I believe that there is reason to doubt that the thermal circuit technique will ever be used to any great extent in calculating either heating loads or cooling loads. Because of its complexity and the lengthy computations which are necessary, the analogy is basically uneconomic under most circumstances. This is not to say, however, that the thermal circuit technique has no place in load calculation. Rather, it is apparent that an indirect application of the thermal circuit must be used if we are to gain from it.

If we are to make maximum use of the thermal circuitry, we must do several things. Among these are a more complete determination of the inputs, resistances, and capacitances for various applications and types of construction. The loads determined by the thermal circuit technique should then be compared with the actual equipment loads in typical installations. Unless this is done, the thermal circuit remains an academic toy rather than a useful tool, since verification of the assumptions used in the analogy is essential.

After verifying the components of the thermal circuit, the factors now used for heat gain and heat loss calculations should be examined. This is the all-important act which

will turn the results of thermal circuit studies into useful tools. It is likely that once we know enough about thermal circuits, the equivalent temperature differentials now given in THE GUIDE can be revised to give heat gains which are more nearly equal to the cooling loads. Similarly, we shall be in a better position to evaluate the transient nature of heat losses and to determine if we might be justified in being somewhat less conservative in our procedures.

It is recognized that this is not a program which can be accomplished over night. It is also recognized that certain simplifying assumptions must be made when the factors for a partially pre-calculated design procedure are developed and that the validity of these assumptions must be checked over a rather wide range of applications. These things take time and yet they are necessary to the transformation of the thermal circuit from a theoretical concept to usable engineering information.

H. B. NOTTAGE, Encino, Calif. (WRITTEN): In order to give practical economic guidance to other persons desiring to solve other similar problems by the methods described in this paper, could there be given a fully explained schedule of the manhours and other direct costs associated with the work reported?

1596

No. 1596

PULSATIONS IN RESIDENTIAL HEATING EQUIPMENT

Preliminary Results

By A. A. PUTNAM* AND W. R. DENNIS**, COLUMBUS, OHIO

This paper is the result of research sponsored jointly by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, the *American Gas Association* and the Oil Heat Institute of America, Inc., conducted through Battelle Memorial Institute.

PULSATIONS have been one of the most annoying problems facing the manufacturer and the distributor of residential heating equipment. A furnace or boiler which *goes into pulsation* will be noisy enough to be heard or felt throughout the dwelling, and sometimes may even cause damage to the house or its contents. Service calls to correct the condition can represent a considerable expense to the organization responsible for satisfactory equipment performance.

To eliminate pulsations, one or more components of the combustion system usually are modified by trial and error until a cure is effected. The modifications may be effective permanently in some installations, and may afford only temporary relief in others. Frequently, the same modifications will not suppress pulsations in other installations apparently similar to the first. Such experiences occur because an understanding of the exact nature and mechanism of the pulsations is lacking.

A research project was activated at Battelle Memorial Institute in May, 1954 to (1) determine the nature of pulsations in oil- and gas-fired residential furnaces and boilers, and (2) find methods for suppressing pulsations. The project is guided by the Pulsations Research Subcommittee of the Technical Advisory Committee on Combustion of the ASHAE.

The chief purposes of the present paper are (1) to discuss theoretical and experimental methods being used to investigate the fundamental nature of pulsations,

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* Assistant Division Chief, Battelle Memorial Institute.
** Principal Physicist, Battelle Memorial Institute.
Presented at the Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Chicago, February, 1957.

and (2) to give an account of the general test results to date. This paper should be considered as a progress report in which are discussed the objectives already attained and the goals still to be reached.

ACOUSTICAL THEORY

Before discussing theoretical considerations and results of tests, certain acoustical terms are defined and certain aspects of standing-wave theory are discussed.

Figs. 1 and 2 show the sinusoidal variation of pressure with time and with position in an open organ pipe, or half-wave tube. Terms which may be clarified by illustrations are denoted on the figures and the Terminology section describes and defines various words and terms used.

Standing Waves: It will be shown later that the oscillations in residential heating equipment studied thus far are of the standing-wave type with one pressure antinode. A brief discussion of standing waves in 4 types of simple systems follows:

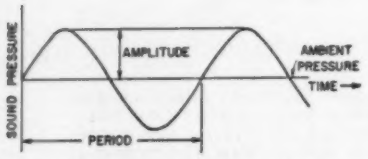


FIG. 1—SINUSOIDAL VARIATION OF PRESSURE WITH TIME

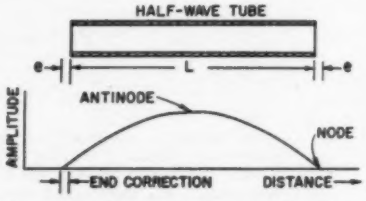


FIG. 2—STANDING WAVE IN AN OPEN ORGAN PIPE (HALF-WAVE TUBE)

The first system to be considered is the open organ pipe, or half-wave tube of Fig. 2. A standing pressure wave in the half-wave tube can be pictured as follows: Periodically, air flows inwardly from the ends, building up a higher pressure region at the center. As the pressure increases, the inflow of air slows down and finally stops. The higher pressure at the center then causes a reversal of flow outwardly toward both ends, accompanied by a lowering of pressure at the center. Because of the momentum of the air, the pressure at the center eventually falls below ambient pressure and with the build-up of this partial vacuum, the out-flow of air finally decreases to zero. Then, the gas starts to rush into the partial vacuum and the cycle repeats. The period, or time for one cycle of pressure variation, is

$$t' = 2(L + 2e)/c = a/c \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where

e = end correction.
c = velocity of sound.
a = wavelength. The length of the tube plus 2 end corrections, $L + 2e$, is equal to $a/2$, so that it is termed a half-wave tube.

Fig. 3 shows the standing pressure wave in a pipe with one end closed. The pressure distribution and the gas motion are the same as for one half of the half-wave tube of Fig. 2. In other words, the half-wave tube could be divided into 2 quarter-wave tubes by a centrally located wall.

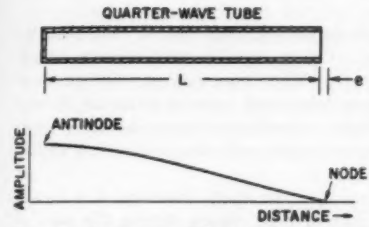


FIG. 3—STANDING WAVE IN A CLOSED ORGAN PIPE (QUARTER-WAVE TUBE)

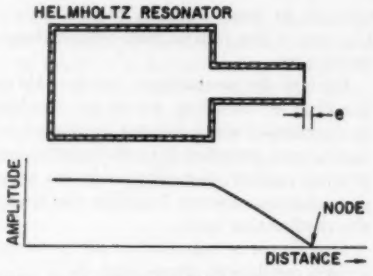


FIG. 4—STANDING WAVE IN A HELMHOLTZ RESONATOR

Fig. 4 shows the standing wave in a cylindrical Helmholtz resonator, which is obtained by enlarging the closed end of the quarter-wave tube. This portion becomes the body, and the open end becomes the neck. If the diameter of the neck is small compared to the diameter of the body, the pressure variation is nearly uniform and at a high amplitude throughout the body, and decreases rapidly along the neck. The gases in the neck act as a mass; those in the body act as a spring. As the neck diameter is increased, the standing wave in the system becomes similar to the standing wave in a quarter-wave tube.

Fig. 5 shows the standing wave in a cylindrical Helmholtz resonator having 2 necks. As the diameters of the necks are increased, the standing wave in the system becomes similar to that in a half-wave tube.

GENERAL PLAN OF RESEARCH PROGRAM

There are several possible mechanisms for generating oscillations in a combustion system. These include: (1) periodic upstream movement of pressure pulses into the supply lines of the burner which may cause periodic changes in the rate of flow of fuel, of air, or of combustible mixture; (2) periodic changes in flame shape; (3) periodic changes in the rate of mass flow into a flame, not associated with upstream movement of pressure pulses; and (4) changes in pressure, which may cause periodic

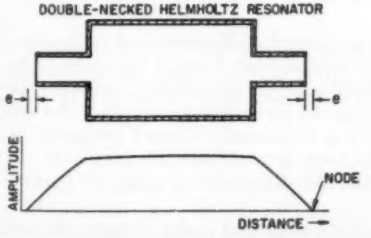


FIG. 5—STANDING WAVE IN A DOUBLE-NECKED HELMHOLTZ RESONATOR

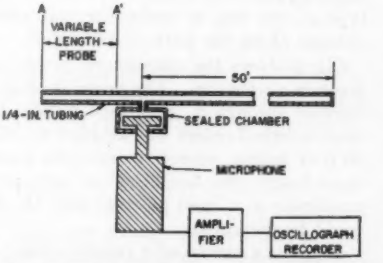


FIG. 6—SKETCH OF MICROPHONE PROBE AND ASSOCIATED EQUIPMENT

changes in mass burning rate. Each of these mechanisms may cause periodic changes in the rate of heat release, from which energy becomes available to drive oscillations.

Each of the mechanisms just listed is usually acoustical in nature, so that acoustic standing or traveling waves are established. Other possible mechanisms should be mentioned which are not acoustical in nature. The flame may amplify pressure oscillations traveling into the combustion zone from such sources as an air blower or even random disturbances in the atmosphere. Oscillations may also arise from disturbances inherent in either the flow pattern or the recirculation pattern within the combustion space.

One factor is common to all these mechanisms. In every instance of a flame-driven oscillation, there must be a greater rate of heat release during the part of each cycle when the pressure is above average than during the part when the pressure is below average. Also, the source of driving energy, or the flame, must be near a pressure antinode.

The oscillations which are driven within particular combustion systems can be divided into 3 broad categories: (1) acoustic standing waves, (2) acoustic traveling waves, and (3) nonacoustic pulses. The first phase of the work was to determine which type was being generated in standard residential heating equipment. The next step was to test 2 conventional furnaces, one gas- and one oil-fired, in the laboratory. In these tests, information as to the mechanism of driving was sought. However, it soon became apparent that the numerous variables involved could not be controlled to a degree sufficient for obtaining accurate data. Therefore, information obtained from tests on the conventional furnaces was used to design 2 experimental furnaces, one gas- and one oil-fired, in which the variables of interest could be more closely controlled. The research is now concerned with these experimental furnaces.

The final phase of the investigation, not yet initiated, will be concerned with devising methods for suppressing the undesirable oscillations. Pertinent fundamental information obtained in the previous phases as to the nature of the oscillations will be applied. In this way, the final objectives can be approached, namely, (1) to furnish service men with means for eliminating oscillations which are consistent with the phenomena which cause them, and (2) to furnish manufacturers with information on which to base designs of oscillation-free equipment.

In the first phase of the investigation, troublesome oscillations were studied in equipment already in operation. Twenty-one units were tested, 10 gas-fired warm-air furnaces, 5 oil-fired warm-air furnaces, and 6 oil-fired boilers. The principal type of test was to probe the unit to determine pressure amplitudes and phase changes along the path of gas travel.

Fig. 6 shows the microphone probe and associated equipment used to measure pressure amplitude. The probe itself is the section from A to A′, which could be inserted through a ¼-in. hole to any desired point in the system. Interference from reflected pulses was avoided by allowing the pressure waves to pass through 50 ft of tubing, wherein most of the energy was dissipated. Using 2 probes of the same length, the amplitude at each probe position could be compared with the amplitude at a fixed position near the flame, and differences in phase at 2 points could be measured.

Fig. 7 is a sketch of a tubeless boiler which was probed in the manner just discussed. Denoted on the sketch are the amplitudes at circled Points 1 through 5 relative to the amplitude at Point 2.

Fig. 8 is a plot of relative amplitude as a function of distance along the path of gas travel for the system of Fig. 7. The curve through the data points resembles the curve shown on Fig. 5 for the standing wave in a double-necked Helmholtz resonator. Similar pressure distributions were encountered in all of the units, both gas- and oil-fired, tested in the field.

No phase changes were detected in any of the units tested in the field; this fact also indicated that each system behaved like a double-necked Helmholtz resonator. The physical ends of a unit were the blast tube and the barometric damper, for oil-fired systems, and the secondary-air inlet and draft hood, for gas-fired systems.

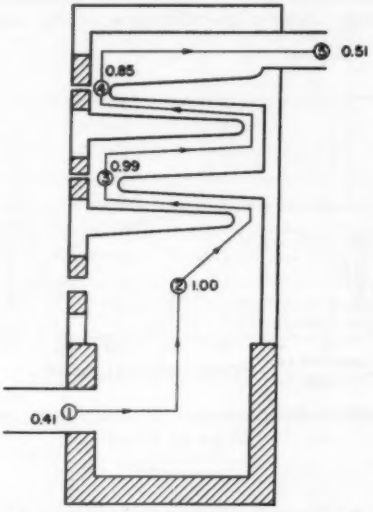


FIG. 7—SKETCH OF TUBELESS BOILER, SHOWING PATH OF GAS TRAVEL (NUMBERS DENOTE AMPLITUDE RELATIVE TO AMPLITUDE AT POINT 2, TAKEN AS 1.00)

It was concluded that the oscillations encountered in the field tests were exclusively of the standing-wave type. Attention was next directed to the flame itself, with the aim of determining the mechanism of energy transfer to the oscillations.

LABORATORY TESTS

The second phase of the investigation constituted a brief test program in the laboratory on 2 conventional furnaces, one gas- and one oil-fired. In preliminary tests, it was found that these units behaved in the same way as units tested in the field. Oscillation studies of a detailed nature were then conducted. The re-

sults of these studies determined what design features should be incorporated in the 2 experimental furnaces which were subsequently constructed and tested. For example, tests on the conventional furnaces showed that provisions must be made for varying under close control the flow rate of combustion air in the oil-fired unit, the flow rates of both primary and secondary air in the gas-fired unit, and the fuel flow rates in both types of furnaces.

The experimental furnaces, one oil-fired and one gas-fired, were designed with 2 chief aims. One aim was to simplify the configuration downstream of the combustion zone to one amenable to acoustical analysis and control. The other aim was to increase the degree of precision with which the most important variables could be controlled and measured. Testing of these experimental units for oscillations is now in progress. After a brief discussion of the configuration of the ex-

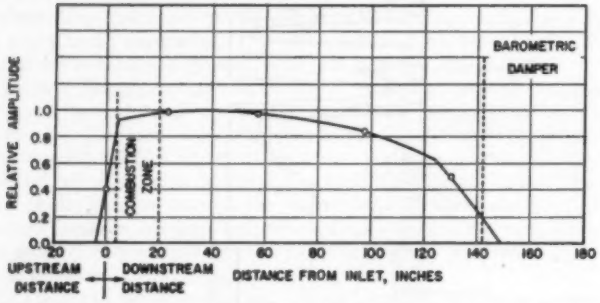


FIG. 8—OSCILLATING-PRESSURE DISTRIBUTION IN TUBELESS BOILER OF FIG. 7

perimental furnaces, the important results obtained thus far are summarized, and samples of the results for 2 types of tests are discussed in detail.

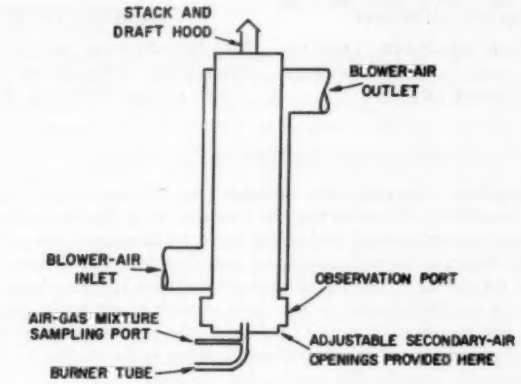
Apparatus: Fig. 9, diagrams (a) and (b), shows gas-fired and oil-fired experimental furnaces, respectively. Each furnace consists of a cylindrical combustion chamber of 14 in. inside diameter, surrounded by an annular duct through which cooling air is circulated. For the gas-fired unit, the combustion chamber is the same diameter throughout, and is 60 in. long. For the oil-fired unit, the first 18 in. is lined with 2 in. of refractory, and the total length of the combustion chamber is about 40 in. Two diametrically opposite observation ports are provided near the inlet end of each furnace. For the oil-fired unit, a high-pressure gun-type burner is used, and for the gas-fired unit, an atmospheric burner of the single-port type is used.

Test Results on the Oil-Fired Unit: In the tests to be discussed, the air first flowed through a metering orifice of the critical-flow type, and then passed through two ¼-in. lines to diametrically opposed points on the blower housing.

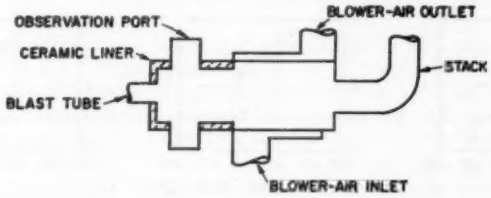
Oscillations in an oil-fired unit are vitally influenced by the interaction of 3 factors, the air-flow pattern, the fuel-spray pattern, and the pattern of recirculation of hot gases in the combustion chamber. This interaction might, under certain

conditions, result in flame instability and periodic variations in rate of burning. The consequent variations in the rate of heat release would furnish the energy to drive the oscillations, and maintain the combination of interactions causing the instability of the flame. Thus, no one of these factors can be isolated and given exclusive responsibility for the promotion of oscillations.

Such an oscillation mechanism is termed autonomous, and is characterized by an oscillating energy source within the system which drives an oscillation to some finite amplitude. At this amplitude, the acoustic damping losses of the system



(a) GAS-FIRED EXPERIMENTAL FURNACE



(b) OIL-FIRED EXPERIMENTAL FURNACE

FIG. 9—SKETCHES OF TWO EXPERIMENTAL FURNACES

balance the amount of energy which is available for driving. The average amplitude remains fairly constant near this value unless suppressive measures are taken.

One way to study the influence of air-flow pattern, recirculation pattern, and fuel-spray pattern was to keep constant the air-flow pattern, and to vary the fuel-spray pattern. This was done by using a number of different fuel nozzles of different capacities, spray angles, and spray types, with no change in blast-tube configuration and therefore no change in air-flow pattern. No effort was made to control the recirculation pattern, which is strongly influenced by the interior configuration of the furnace.

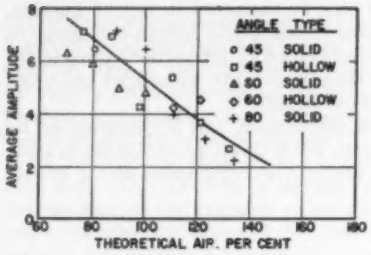


FIG. 10*a*—HIGH-AMPLITUDE OSCILLATION DATA FOR 1.0-gph NOZZLES, SPARK OFF

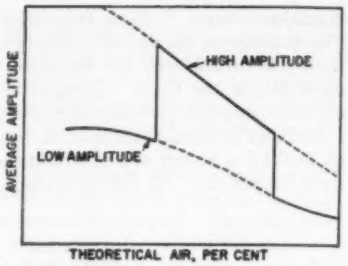


FIG. 10*b*—SKETCH OF LOW- AND HIGH-AMPLITUDE RANGES OF OSCILLATION FOR 1.0-gph NOZZLES, SPARK OFF

There were two results of particular interest, as follows.

(1) The amplitude of oscillation increased as the opening of the barometric damper was increased. Considering the furnace as a double-necked resonator, with the damper as the physical end of one neck, an increase in area of the damper opening would decrease the end correction, and therefore would decrease the effective length of this neck. This, in turn, would decrease the wavelength of the system and also move the location of the cross-section of maximum pressure amplitude toward the combustion zone. Bringing the region of maximum pressure amplitude and the combustion zone closer together is conducive to more efficient driving of the oscillations by the combustion process.

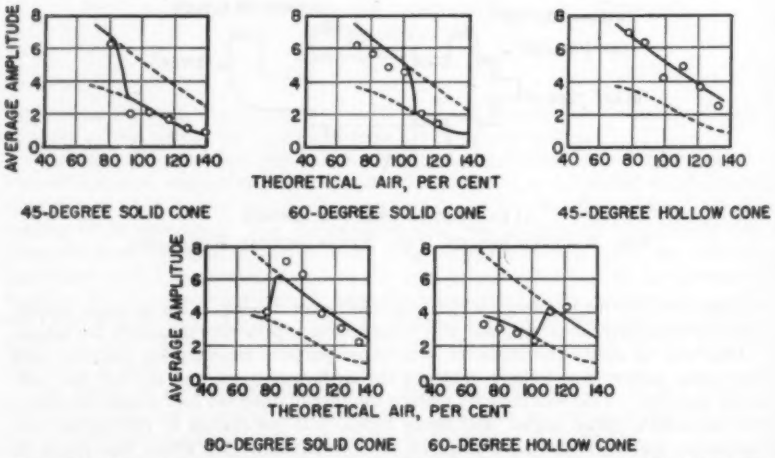


FIG. 11—HIGH- AND LOW-AMPLITUDE OSCILLATION RANGES FOR 1.0-gph NOZZLES OF VARIOUS SPRAY ANGLES AND SPRAY TYPES, SPARK OFF

(2) The amplitude of oscillation was considerably higher with the spark turned off than with the spark left continuously. Apparently, continuous ignition in the experimental furnace had a stabilizing effect on the flame which reduced the oscillating component of the rate of heat release.

The amplitude data for all nozzles at one setting of the barometric damper, and with spark off, plotted as a function of air-fuel ratio fell into 2 distinct classes, high amplitudes and low amplitudes. Within each class, the amplitude decreased with increase in percentage theoretical air. The high-amplitude type was loud, and would be extremely troublesome in practice. The low-amplitude type was loud at rich mixture, but decreased in intensity to that of normal flame noise as the percentage of theoretical air was increased.

Fig. 10*a* shows how the high amplitude curve between 70 and 140 percent theoretical air was obtained. The low-amplitude curve was obtained similarly from the low amplitude data. Fig. 10*b* is a sketch of the high- and low-amplitude curves. A typical curve through the data for a particular nozzle is indicated by the solid curve. The location of the high-amplitude section, in terms of mixture composition, varied from nozzle to nozzle. The tests were extended to a far richer mixture ratio than would be used in practice to permit the establishment and definition of such general results as are presented herein.

Fig. 11 shows how the high- and low-amplitude sections of the curves through the data shifted with change in spray angle and spray type for the 1.0-gph nozzles used. The range of mixture ratios covered was not broad enough to show the entire high-amplitude range, as drawn in Fig. 10*b*. The curves are the average curves for all the data, including nozzle capacities from 0.75 to 1.25 gph, but excluding 80- and 90-deg hollow-cone nozzles. The high- and low-amplitude curves are connected in the region between 2 data points where the oscillation changed from one type to the other. In each plot, as in Fig. 10*b*, the portions of each curve for which oscillations of that particular type were generated are shown as solid lines.

The plots are ordered as follows, reading from left to right and starting at the upper left; solid cone, 45 deg; solid cone, 60 deg; hollow cone, 45 deg; solid cone, 80 deg; hollow cone, 60 deg. The order shows consistent changes in the positions of the high- and low-amplitude ranges. The order of the nozzles in terms of their acoustical behavior was found to be the same as their order in terms of calculated effective spray angle.

For the 80- and 90-deg hollow-cone nozzles, which were omitted in the original analysis of the data, the curve for the low-amplitude oscillations was distorted somewhat on the rich side. Otherwise the results conformed to those already discussed.

The results just discussed were obtained by varying the type of spray nozzle without any change in the pattern of air flow around the nozzle. It is expected that similar results will be obtained by changing the air distribution, using a fixed spray.

Motion-picture studies have also been of value in studying oscillations in the oil-fired unit. They have been used to identify characteristics of the visible combustion process which might be related to the mechanism of driving.

Fig. 12 is a typical sequence of frames taken from high-speed motion pictures of the flame in the experimental oil-fired furnace. The film speed was approximately 900 frames per second for these photographs. Flow is from right to left,

with the fuel-spray nozzle just out of view to the right at the center of the frame. Referring to Fig. 9, the photographs were taken through the lower view port, with the upper port covered. The field of view is about 7 in. along the axis of the combustion chamber.

The distinguishing characteristic of the combustion is the extremely small *dot* of flame somewhat below the axis and near the midpoint of the first frame. This

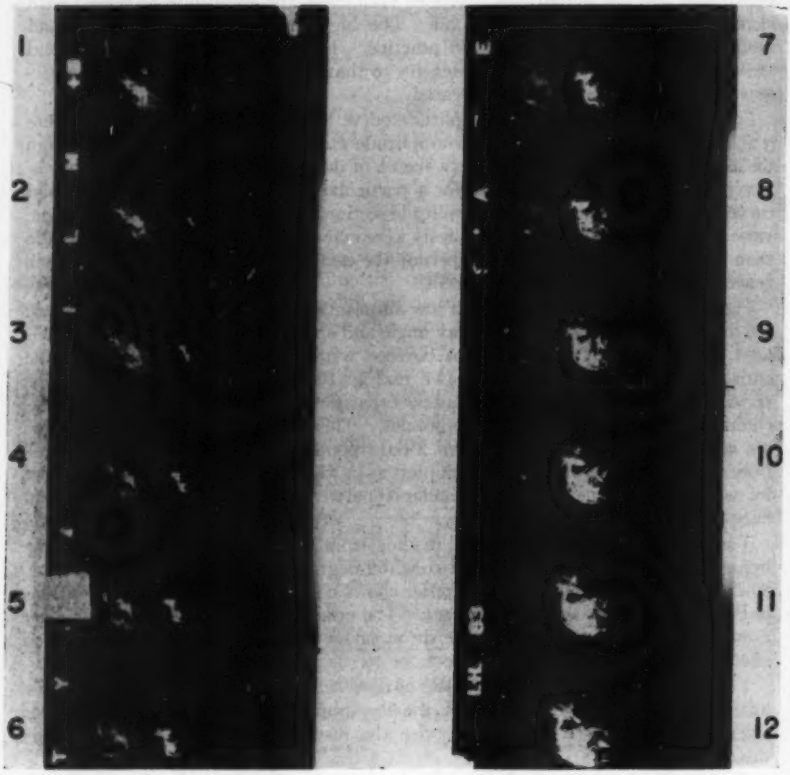


FIG. 12—HIGH-SPEED PHOTOGRAPHS OF OSCILLATING OIL FLAME

small area of flame expands rapidly from frame to frame, and has become large in Frame 12. Simultaneously, the large area of flame at the extreme left of Frame 1 rapidly moves out of view and has almost completely disappeared in Frame 12. This flame apparently was the remnant of the flame which expanded immediately before the one just described.

The flame expansion seen in the 12 frames of Fig. 12 can be considered as the beginning of one cycle of oscillation, and covers about ⅓ of a complete cycle.

During the remainder of the cycle, the flame continues to cover a large area and moves downstream to the left, finally fading out as did the previous flame seen at the left of Frame 1. The expansion of a small *dot* into a large flame occurred once per cycle. This fact led to the tentative conclusion that the sudden release of energy from the rapid burning is responsible for driving the observed oscillations.

Test Results on the Gas-Fired Unit: The test results for the experimental gas-fired single-port furnace can be summarized as follows. For a given furnace configuration, the effects of (*a*) fuel-flow rate, (*b*) primary air-flow rate, and (*c*) secondary air-flow rate and distribution on generation of oscillations were studied. As the fuel flow rate was increased, the amplitude of oscillation increased. This was apparently a result of the additional energy available for driving oscillations as a rate of combustion was increased. Secondary air-flow rate and distribution have shown only a slight effect on oscillations in the experiments made thus far, and do not appear to be of primary importance.

At the present stage of the investigation, the rate of flow of primary air in a given burner configuration appears to be the most significant variable affecting generation of oscillations. In a series of tests in which the primary air-flow rate was varied, while keeping the fuel-flow rate and secondary air-flow rate constant, the phenomenon similar to that just described for the oil-fired unit was observed. This was the appearance of both a low-amplitude and a high-amplitude type of oscillation. The phenomenon differs from that of the oil-fired unit in that no changes in operating conditions were made during the time that the high-amplitude oscillations appeared. At low primary air-flow rates, no high-amplitude oscillations were generated. As the primary air-flow rate was increased, high-amplitude oscillations were generated in short bursts for increasing percentages of a fixed time interval of operation. However, at no time were high-amplitude oscillations generated exclusively.

The 2 types of gas-furnace oscillations encountered were distinguished not only by difference in amplitude, but also by the appearance of the flame in each case. In observing the flame around a 7-in.-diameter flame spreader located at the rather close distance of 1 in. above the burner port, the flame appeared as a bluish ring about ½ in. inside the edge of the spreader when the low-amplitude oscillations were generated. This flame appeared stable. When the high-amplitude oscillations occurred, the entire under surface of the spreader appeared to be speckled with shortlived blue flames.

The dependence of the time at high amplitude on changes in the configuration of the furnace, such as changes in length of primary air duct and height of spreader above the duct, has been demonstrated. A detailed investigation of these effects is expected to show that for some conditions, high-amplitude oscillations will occur over the entire range of reasonable primary air-fuel ratios.

Fig. 13 shows a sequence of frames from a high-speed motion picture of the gas flame, colored with sodium, taken during high-amplitude oscillations, covering about one cycle of oscillation. Owing to the high frame speed of about 850 cps, only alternate frames are shown herein. The horizontal field of view was slightly greater than 7 in. wide at the axis of the combustion chamber. Flow is from bottom to top on each frame. A small portion of the flame spreader, which is intensely illuminated by the flame underneath, is cut off at the extreme left hand side of the frame.

In Frame 1 of Fig. 13, the near edge of a vortex ring resembling the well-known smoke ring, is just appearing at the top front edge of the spreader. In Frame 4,

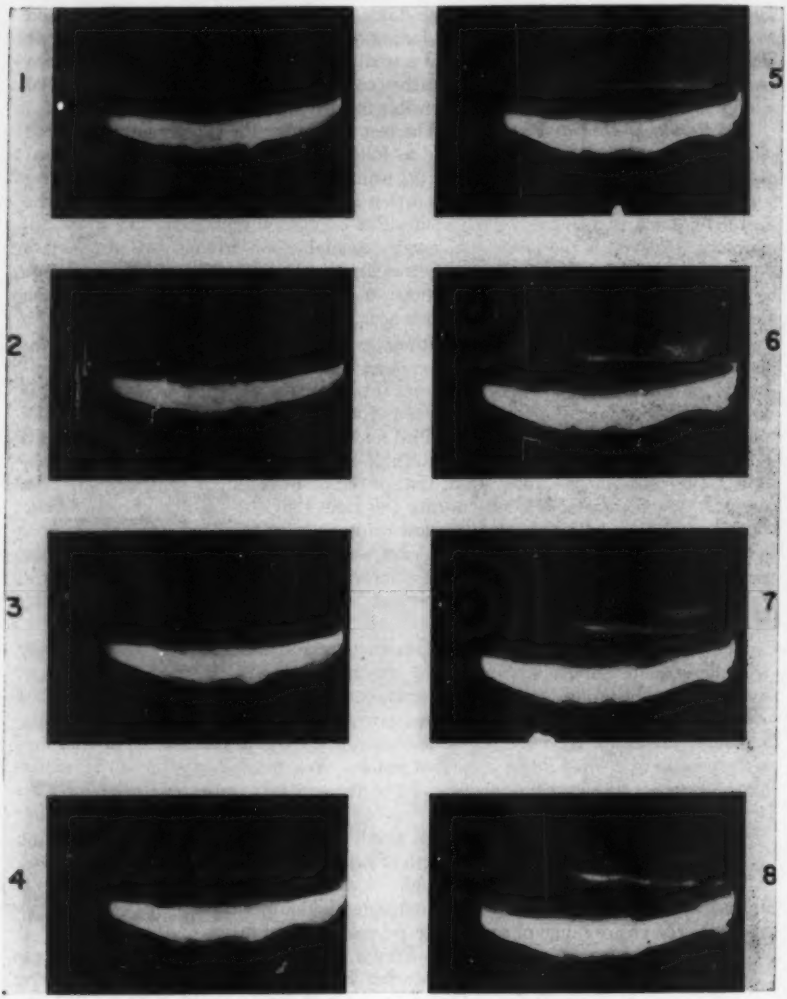


FIG. 13—HIGH-SPEED PHOTOGRAPHS OF OSCILLATING GAS FLAME

the vortex ring has risen an appreciable distance, but as a result of the viewing angle, the back edge of the ring is just coming into view, although the vortex is horizontal. In Frame 8, the entire ring, which is by now somewhat distorted and broken, is above the spreader, and is being viewed from underneath. Evidence

that the vortex is originally generated at the lip of the burner tube is given by the progressive movement of a distortion up the underside of the flame surface. This distortion is quite clear on the right side beginning with Frame 2 and continuing through Frame 8.

Many examples are known of the phenomena of vortex shedding. For instance, an oscillating gas column in a tube can shed vortices from the edge of the tube. A jet of gas impinging on a wedge at the proper location will also produce vortices. Usually, neither of these phenomena reaches an amplitude high enough to produce audible sound. However, if a natural shedding time and one of the natural periods of the tube agree closely, the vortex shedding and the natural oscillation of the gases in the tube can reinforce each other, and high-amplitude oscillations may result. Such a phenomenon may be occurring in the gas furnace being studied, with an additional factor of periodic energy addition by a pulsating heat-release rate.

FUTURE WORK

The next phase of the experimental work on the oil-fired experimental furnace will cover the effects of air-flow pattern on oscillations, using a fixed spray pattern. At the conclusion of these tests, it is expected that sufficient information on the mechanism of driving will be available to permit a study of methods of eliminating or suppressing the oscillations.

Future work on the gas-fired experimental furnace will be concerned with the effects of primary air-flow rate, primary air-fuel ratio, and burner-tube configuration on generation of high-amplitude oscillations. Motion-picture studies will be continued and model studies are planned to obtain further information on the vortex-shedding phenomenon. The flow characteristics within the burner tube are also considered of importance, and will be investigated.

ACKNOWLEDGMENT

The able assistance of the members of the Pulsation Research Steering Subcommittee of the TAC on Combustion of ASHAE in guiding the research and providing necessary information and materials is gratefully acknowledged. Thanks are due to the organizations and individuals, too numerous to mention, whose cooperation was essential to the successful completion of the field work. Special thanks are due to the personnel of the Armstrong Furnace Company, Columbus, Ohio, who provided the proving ground for the first stages of the field work, and who also constructed the experimental furnaces.

TERMINOLOGY

1. Oscillation—periodic variation, with time, of pressure with respect to the ambient pressure. During oscillation, the pressure is alternately greater than and less than the ambient pressure.
2. Pulsation—here considered to be the observable effects of the pressure oscillations. That is, pulsations can be seen, felt, or heard by an observer, or measured as changes in pressure of the surrounding atmosphere or as forces exerted on surrounding surfaces. Since there are certain physiological implications connected with the term *pulsation*, the term *oscillation* is used in the subsequent discussion.
3. Cycle—complete sequence of values of pressure; at the end of a cycle, the pressure and the rate of change of pressure together repeat themselves. Note that the pressure

assumes the same value at 2 times during a cycle, unless the cycle is taken from one time of maximum or minimum pressure to the next.

4. Period—time interval consumed by one cycle.

5. Frequency—number of periods occurring in one second, or number of cycles per second (cps).

6. Amplitude—absolute value of the maximum difference between the instantaneous static pressure and the ambient pressure in a given period.

7. Phase—fractional part of a period through which the independent variable, usually time, has advanced, measured from an arbitrary origin. The origin is usually chosen so that the fraction is less than unity.

8. Wavelength—distance between 2 wave fronts in which the pressures have a difference in phase of one complete period. During one period, the wave has advanced a distance equal to the product of the velocity of sound and the period.

9. Standing Wave—oscillations having a fixed distribution of amplitudes in space, and characterized by the existence of nodes and antinodes that are fixed in space.

10. Nodes—points, lines, or surfaces in a standing wave where there is zero amplitude.

11. Antinodes—points, lines, or surfaces in a standing wave where there is maximum amplitude.

12. End Correction—length added to an oscillating gaseous system to account for the fact that the pressure node at an open end is located somewhat beyond the physical end of the system. The size of the correction depends upon the configuration of the opening. For a cylindrical pipe, the correction is usually taken as 0.6 of the radius.

DISCUSSION

W. B. KIRK, Cleveland, Ohio, (WRITTEN): In their preliminary report on pulsations in residential heating equipment, the authors have done well in setting forth and describing elements of the physical theory applicable to this problem. These explanations and terminology definitions should prove quite helpful to those interested in the subject.

That portion of the paper dealing with gas-fired units establishes primary air as a major factor influencing the occurrence of oscillations, at least with single port target-type burners. It would have been helpful had a sample plot of data, or the span of data for a typical experiment, been included for review in conjunction with the vortex-shedding theory. It seems possible that the conclusion regarding primary air effects might be modified at some air-gas mixture where the effects of secondary air become predominant.

Although not specifically mentioned for future work, it might be well to consider transient effects of temperature and burning speed of the fuel gas used. For example, it has been observed in practice that some furnaces will pulsate when *cold* but not after they have become heated. The reverse has also been observed, in that a furnace may not pulsate until after it has become heated. Thus, in establishing standing-wave patterns for a furnace, temperature may become an important factor. This thinking might also apply to oil-fired furnaces.

Since preparation of this paper, Mr. Putnam has found that oscillations occurring with multiple-port gas burners apparently have a different acoustic driving mechanism from that of single port target-type burners. Future reports on this phase of the work, in particular, should therefore be of more than usual interest because of the wide-spread use of multiple port burners in residential heating equipment.

Other future work mentioned in the paper is highly desirable. Upon completion of studies with the experimental furnaces, it is hoped that the project will include test application of the findings to at least one typical furnace or boiler using the types of burners studied.

H. R. HEIPLE*, Sewaren, N. J., (WRITTEN): The authors have done a commendable job in presenting this preliminary report of their study of pulsations in residential heating

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* Technical Division, Oil Heat Institute of America.

equipment. Inclusion of a brief discussion of acoustical theory has helped materially in making the paper understandable to laymen. Presentation of the general plan of the research program early in the paper lets the reader determine the progress already made. The paper is, accordingly, well organized and presented.

The discovery that all the installations examined, both in the field and in the laboratory, oscillated in the same manner, that is, as double-necked Helmholtz resonators, is of interest and of value. It has reduced the problem to a study of the factors involved in flame-driven oscillations in heating equipment.

Some readers may regard the use of experimental furnaces in this investigation with distrust. The authors have clearly indicated their reasons for building these units and conducting tests thereon. The results obtained show the wisdom of their decision.

The writer feels too much emphasis has been placed on the role of the oil spray pattern in pulsation of oil-fired equipment. It is true that the authors state that oil-spray pattern, air-flow pattern, and the pattern of recirculation of hot gases interact to promote pulsations, but the emphasis, textually, is on oil-spray pattern. The authors indicate their belief that the air-flow pattern did not change in these tests. The writer questions whether the air-flow pattern actually remained constant while the quantity of air supplied was varied from 70 to 140 percent of theoretical.

That high amplitude and low amplitude oscillations are two distinct *types* of oscillation, as the authors apparently infer, has not been established. Were differences other than amplitude observed between these two *types* of oscillation? Can the occurrence of two such *types* of oscillation be explained?

The method used to denote mixture composition, i.e., in terms of percent of theoretical air, may be unintelligible to a large number of prospective readers of this paper. Most oil burner people use *percent CO_2 in the flue gas* or *percent excess air* as a measure of air-fuel ratio. A supplementary scale giving these values would help these people understand the plotted data.

The range of compositions in which tests were made are not realistic in relation to the air-fuel ratios generally used in oil-burning equipment. The authors have indicated that rich mixture ratios were used to establish and define a general theory. It is now necessary to confirm the theory in the range of compositions used in the field, that is, from about 120 to 250 percent of theoretical air (about 12.8 percent CO_2 to 6 percent CO_2).

The writer does not feel qualified to discuss the section of the paper relating to studies in gas-fired equipment. It is of interest to note, however, that the high-speed photographs show the mechanisms by which oscillations are generated in gas-fired and oil-fired equipment to be markedly different. This is not unexpected, since the processes involved in burning these two fuels differ radically.

The paper contains, as was indicated above, much material of interest. It is hoped that papers dealing with future developments in this investigation will be equally interesting, and that one of these papers will give practical answers to the perplexing problem of the elimination of pulsation in residential heating equipment.

A. A. MARKS, Philadelphia, Penn., (WRITTEN): The solution of the problem of pulsation and resonance in heating equipment is of considerable interest to the heating industry. The research program on this subject, conducted by Battelle Memorial Institute, is sponsored jointly by ASHAE, AGA and OHI.

The objective of this project is to determine the nature of pulsation and resonance in oil and gas fired heating equipment and to find methods for eliminating or suppressing such oscillations.

The paper presented at this meeting is one of a series of papers to follow, covering the progress of the research program. A second paper on the *Acoustic Coupling of Residential Furnaces and Their Surroundings* has been submitted to the Society for publication, and will probably be presented at the summer meeting of the Society.

By the end of the project year, May 31, 1957, a final report on the Mechanism of Oscillation in Oil and Gas Fired Equipment will be made available, and some methods

for the suppression and elimination of these oscillations will be proposed. Not all of the answers will be available at that time, and it is proposed by the Steering Committee of the project and by the TAC on Combustion to extend this project for another year. This will allow more time to elaborate and clarify the theory of the Mechanism of Oscillation, and to extensively study possible means for suppressing the oscillations.

The authors of this paper are to be congratulated on the technical excellence with which resulting test data has been presented, and it is hoped that this work will be extended to effect an enlarged understanding of the problem and its suppression.

BÉLA KARLOVITZ†, Pittsburgh, Penn., (WRITTEN): The work described in this paper is concerned with the difficult problem of flame generated oscillations in practical combustion equipment. The work is organized according to an adequate general plan of research which includes preliminary measurements on practical heating equipment, theoretical studies and laboratory tests on experimental combustion chambers. The preliminary measurements clearly demonstrate the role of acoustical standing wave oscillations in the combustion chamber for the generation of pulsations. The more difficult part of the problem is to find the mechanism by which the flames can drive these acoustical oscillations. The paper describes 4 different possible mechanisms which may be responsible for the excitation of the oscillations. One important possible mechanism is not mentioned among these possibilities. This is the periodical extinction and reignition of the flame. This can play an important role if flame stabilization is marginal. The importance of this mechanism is indicated in the paper by the experience that continuous operation of the ignition spark substantially reduced the amplitude of oscillations in the case of oil fired furnaces.

I believe that the authors attacked the problem on a sufficiently broad fundamental basis to reach valid conclusions. The continuation of this work is expected to lead to valuable results in our understanding of flame generated oscillations, and to the elimination of oscillatory disturbances from practical installations.

G. J. SANDERS‡, Chicago, Illinois: The Foundation has been carrying on a program very similar to the one discussed by Mr. Putnam. The object of our program was to try to find practical applications of combustion noise theory so that production furnaces would be up to date with all the modern technology. This involved quite a bit of basic research into the phenomena and many of our findings are very similar to those given by Mr. Putnam.

Yesterday, Mr. Putnam and several members of our staff spent the whole day discussing combustion. Our findings include a few things which may add to the theory developed by Mr. Putnam. In our work, instead of looking at an oil furnace flame in the form of two doughnuts outside of a fuel spray, we have considered it as a solid cone of burning inside the fuel spray and a hollow cone of burning outside the fuel spray. This is just a slightly different conception but it is based on the same type of analysis.

Actually we find that these 2 types of burning can be separated and we can get either one independently with the proper type of installation. When this has been accomplished it is relatively easy to learn more about what causes and what changes each of these 2 separate types of combustion.

We also have some high-speed movies which tend to indicate that perhaps the basic phenomena behind the oil burner problem is very similar to that behind the gas problem. In other words, we have movies which show a semblance of a vortex shedding. In the gas furnace there is a spreader and shedding apparently starts at the nozzle and goes about the spreader. In the oil furnace the cone of the fuel squirting into the combustion chamber performs very similar to a type of spreader and the shedding seems to take place somewhere about the cone of the fuel rather than about a mechanical object.

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† Combustion and Explosives Research, Inc.
‡ Armour Research Foundation of Illinois Institute of Technology.

Another important point agrees with Mr. Putnam's theory. It was mentioned that the oscillations are flame driven. We find some oscillations are flame driven but we also find other oscillations which are flame amplified rather than flame driven.

For example, if there is a source of turbulence before the mixing, this will be carried on into the process and produce rough combustion. In this case the flame merely amplifies the roughness which is already in either the fuel or the air stream.

AUTHORS' CLOSURE (Mr. Putnam): The interest shown in the paper is appreciated by the authors, and it is desired to thank each of the discussors for his remarks. These remarks significantly increase the value of the paper.

Mr. Marks has acted as the chairman of the Pulsation Steering Committee for two years, and in that time has lead the group ably. He, and the other members of the committee, have been of inestimable help in contributing their long background of practical experience, aiding in the design of experimental equipment and test procedures, and obtaining items necessary for the conduct of the program. It has been a pleasure to work with Mr. Marks and his committee.

Mr. Kirk has raised some interesting questions concerning the difference between the mechanism of oscillation production in single-port and multiple-port gas furnaces, the effect of secondary air, and the effects observed in cold start. As indicated in the paper, the mechanism associated with the single-port target-type burner is related to a vortex shedding phenomenon; recently obtained evidence confirms this point. The periodic vortices cause a periodic disturbance of the flame which is rather definite near the burner but becomes more variable farther downstream. This may be noted to some extent in Fig. 13. It follows that the most important part of the flame, as far as driving the oscillation is concerned, is that nearest the burner port. In this region the supply of secondary air is adequate unless the over-all supply of secondary air is too low, or special construction features are present which force recirculated products of combustion into this region. This explains why the tests in the experimental furnace are not critical with respect to the supply of secondary air. In the case of the multiple-port burner, a different mechanism of driving, not involving vortex shedding, is present, as will be discussed in a future paper. However, the same remarks hold for the multiple-port burner as for the single-port burner insofar as the effect of secondary air is concerned. In a production-type furnace, it should be noticed that secondary air is usually changed by making a change in an opening, which also changes the acoustic properties of the furnace. This change in acoustic properties can affect the noise output.

The question of the cause of difference in noise output between cold start and normal operation in many furnaces is difficult to answer without referring to a specific furnace. The reason is that several changes take place when going from cold start to normal operation which can affect the acoustic action of the furnace. The most obvious is that the natural acoustic frequency of the furnace will change as the temperature of the gases within it increase. Another factor is that the burner heats up, causing the viscosity of the gaseous primary mixture to increase. This decreases the air flow rate, and as a result alters the primary mixture ratio, the flame speed of the primary mixtures, the heat-release rate in the region that affects acoustic driving, and the Reynolds number. All these factors influence the acoustic output, but without specific details of design one cannot evaluate the relative effect of each change, and thus which change is most directly the cause of the change in noise output from cold start to normal operation.

Mr. Karlowitz has commented on the extinction and reignition process as a source of driving energy to the oscillation of the oil furnace. That this is the case is confirmed by high-speed photographs such as in Fig. 12. It is the authors' opinion that such a phenomenon may be classified as a severe example of driving by periodic changes in flame shape. However, a more understandable explanation of the driving mechanism results from consideration of the phenomenon directly as one of periodic extinction and reignition, as Mr. Karlowitz suggests. The authors believe the effect of the spark in decreasing the amplitude of oscillation is twofold; it stabilizes the flame and it consumes in the spark trail a potential supply of driving energy.

Dr. Heiple covers several points of interest in connection with the tests of the oil furnaces. As he states, the emphasis in this paper is on the oil-spray pattern; this results from the fact that oil spray was the most easily controlled variable and thus was chosen as the first to study. Later tests cover the effect of air pattern. With some blast tubes, the air pattern seems relatively independent of the air-flow rate, while with others, air pattern changes considerably with air-flow rate. Furthermore, the air pattern can affect the oil-spray pattern. Since the acoustic noise is produced by the interaction of air pattern, oil-spray pattern, and pattern of flow of the recirculating products of combustion, the change of all three with change in operating conditions must be understood to explain the acoustic results.

In reference to the 2 types of oscillations, it is sufficient for the present discussion to note that in the high-amplitude oscillation, all occurrences appear to be timed correctly relative to each other to drive the oscillations; because of the sudden jumps in amplitude which are observed, it is felt that this timing may be connected with a *lock-in* process. For the low-amplitude situation, the oscillation may get started but then the timing of the various occurrences do not remain in the correct relation, or other sources of driving energy also are produced that tend to counteract the first, or both. A more detailed discussion of the mechanism will follow in a future paper after certain points are clarified by definitive tests.

As Dr. Heiple points out, the range of composition studied is wider and richer than encountered in normal operation. The reason is that for the particular combination of blast tube and furnace discussed herein, the phenomena which are of interest occurred at high amplitude in this range. The range was made as wide as possible to obtain sufficient information to form a basis for explaining the phenomena. Later data on other blast tubes have shown that the observed phenomenon is general, but the jump points change with change in air-distribution patterns.

Dr. Sanders remarks about the flame *donuts* require some clarification, since they are not covered in the report. The high-speed photographs taken by the authors indicated 2 regions of periodic ignition which may be considered as donuts symmetric to the burner axis. One donut lies a few inches downstream of the nozzle and close to the axis. The ignition shown in Frame 1 of Fig. 12 occurs in this donut. A second donut of potential ignition sources is larger in diameter and closer to the blast tube. It lies outside the spray cone. Dr. Sanders considers the steady-state process and thus the cones of flame extending downstream from such donut-like regions.

Just as it is possible for periodic ignition to take place from either or both of the regions, it is possible to have a steady-state flame propagating from either or both regions. In the case of the outer region, Dr. Sanders considers the recirculation zone of hot products of combustion as a potential shed vortex, which will shed and reform under oscillating conditions. This concept would indicate a similarity between the phenomenon in the oil furnace in this region and the phenomenon in the single-port furnace. It is conceivable that in some instances the periodically ignited flame in this region moves downstream as it expands while in other instances, vortices shed when a periodic oscillation occurs in the furnace. Further information will have to be obtained on this point before it can be completely resolved.

Dr. Sanders' mention of the possibility of flame amplification of noise is greatly appreciated. Such amplification has been discussed in the literature for certain experimental combustion systems, and the flame amplification of spark noise is a well recognized phenomenon. The authors agree with Dr. Sanders that such action may also take place in the low-amplitude region of oscillation. Thus, improperly designed fans, and unnecessary grids, and obstructions in the blast tube or before it, could lead to a more noisy furnace.

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No. 1597

THE DUST SPOT METHOD FOR EVALUATING AIR CLEANERS

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This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, in cooperation with the Mechanical Engineering Department, University of Minnesota, Minneapolis, Minn.

TESTING of air cleaners for effectiveness requires the sampling and evaluation of the airborne dust concentration before and after the air cleaner. This may be accomplished in a number of ways, the meaning of the resulting efficiency number depending on the dust property evaluated. The 3 most common methods of sample evaluation are by counting the number of particles, by comparison of the optical quality of samples collected on suitable media and by weighing.

Each of these has its sphere of usefulness. If the test dust is homogeneous with respect to particle size, then all 3 methods will yield equivalent efficiency values, but if the dust has a relatively broad distribution, as is usually the case, then the 3 evaluations will be quite different. For example, a conventional permanent type viscous filter that was investigated had an efficiency on normal air-borne dust of 53 percent by weight, 14 percent by dust spot and 0.4 percent by number.

OPTICAL METHODS

A number of different optical methods for evaluating air cleaners have been developed. Perhaps the best known of these is the so-called dust spot method developed by the Bureau of Standards[1]. In this method, the dust laden air is drawn through a filter paper medium and the resulting spot is evaluated by visual comparison or by photometric measurement of the light transmitted or reflected by the paper.

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[1] Exponent numerals refer to References.

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During and after World War II, several instruments capable of measuring the light scattered from the air-borne particles in situ were developed under government sponsorship. In the penetrometer type instrument[3], an intense light beam is passed through the dusty air and the light scattered from the dust particles is measured by a sensitive photo tube. The relative amount of light scattered before and after the cleaner is an index of the air cleaner efficiency. These instruments have been used extensively in combination with homogeneous aerosol generators for the evaluation of high efficiency air cleaners.

Another type[3, 4] counts and measures the magnitude of the light flashes as each particle passes through the light beam. With this instrument, an estimate of the size distribution may be obtained in addition to the concentration. However, these counters are very costly.

THE DUST SPOT METHOD

Previous Work: Because of its simplicity and convenience, the dust spot method has been widely used. A variety of filter media and measuring techniques have been used. Among the better known are variations of the method developed by Dill at the National Bureau of Standards[1]. In this method the upstream and downstream samples of dust collected on chemical filter paper are evaluated in a simple photometer. To eliminate the need for an accurate knowledge of the relationship between optical density and dust concentration, the air-flow rate and sampling time is adjusted so that the 2 samples have approximately the same optical density.

A later instrument used with the Dill dust spot method combines the filter paper holders and photometer into a single unit for convenience and portability. Here the 2 filter papers and photocells are placed on opposite sides of a lamp, and so connected to a galvanometer that an unequal rate of decrease of the light transmission of the 2 filters is indicated by the galvanometer deflection. The relative air flow through the 2 filters at balance is a measure of filter efficiency.

Most of the dust spot methods have used the reduction in light transmission of a translucent filter as a measure of the dust deposit. Studies by Rowley and Jordan[5] have shown that the reflected light is not as reliable a measure of the dust deposit as is the transmitted light.

Present Research at the University of Minnesota: As part of the research on air cleaning, a preliminary study of the dust spot method was made. This study showed that such factors as optical design of the photometer and filter media used could have a considerable bearing on the resulting measurements. Accordingly, research was initiated to further study the relative performance of available filter media and to develop an optimum photometer design. At the same time, a study was made to discover exactly what dust properties were being measured.

Theory: Because theoretical basis for the dust spot method has been discussed in detail elsewhere[5, 6, 7], only a brief outline is given here.

The basic equation for calculating the efficiency of an air cleaner from upstream and downstream samples is given by:

$$\eta = 1 - [(C_d/C_u)(q_u/q_d)(t_u/t_d)(A_u/A_d)] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

The dust spot method may be used with Equation 1 to evaluate air cleaners in several ways. If the exact relationship between C and the reduction in light transmission due to the dust spot is not known, the other variables may be selected so that the optical quality of the upstream and downstream samples are equal and

$C_d = C_u$. Though this procedure eliminates the need for knowing the *OD**-*C* relationship, it has disadvantages.

The ratio A_u/A_d is usually fixed by the sampler design, often at a 1 to 1 ratio. Also the practical limitations of flowmeters in combination with the pressure drop characteristics of high efficiency sampling media usually make it impractical to obtain more than a 1 to 10 ratio of upstream to downstream air-flow rates. The use of variable sampling times requires that the dust feed rate or dust concentration to the air cleaner be constant. However, the greatest disadvantage stems from the additional effort involved in adjusting sampling areas, air flows or sampling

TABLE 1—PROPERTIES OF SAMPLING MEDIA

PROPERTY	CHEMICAL FILTER PAPER W-41	GLASS FILTER PAPER	AA MILLIPORE
1 Strength	Good	Fair—Requires Support	Fragile—Requires Support
2 Light transmission, in % ($\Theta = 26°$)	5.2	1.2	0.8
3 Flow p_0	0.013	0.043	0.24
Coefficients[a] n	1.05	1.04	0.82
4 Lambert-Beer Law	Fair	Good	Good
5 Dust spot arrestance on atmospheric dust	50–95%	99+	100%
6 Uniformity	Fair	Subject to pinholes	Good	Good
7 *OD* Atmospheric dust	1.2–1.3	1.2	1.0
OD (millipore) Carbon Dust	1.2	1.0	1.0
8 Suitability for weighing of dust on filter	Fair	Excellent	Fair
9 Suitability for microscope of Sedimentation size analysis	No	No	Good

[a] Flow Coefficients: $p = p_0 V^n$, with V in ft/min; and p_0 in cm of Hg.

times to obtain equal optical densities upstream and downstream while still maintaining isokinetic flow through the sampler heads.

For these reasons it appeared desirable to study the *OD-C* relationship to discover the magnitude of the deviation from the Lambert-Beer law and to define the possible conditions under which variable *OD* might be used in air cleaner testing.

SAMPLING MEDIA

Because of their relatively high light transmission, strength and availability chemical filter papers have been most frequently used as dust spot sampling media. Within recent years higher efficiency glass fiber papers and millipore filters have become available. In view of certain known deficiencies of the chemical papers, a study of all 3 was made to determine the properties of each.

Important properties are given in Table 1. The pressure drop coefficients given are valid up to a face velocity of about 150 fpm. Above 150 fpm, n slowly in-

* Throughout the paper *OD* is used to represent optical density.

creases. The dust spot arrestance of the W-41 paper was measured at a face velocity of 40 fpm using a millipore filter downstream of the paper. A series of such measurements under different weather conditions in the University of Minnesota Particle Laboratory on different samples of paper gave arrestance values from 43 to 95 percent with the average being about 88 percent. The 43 percent arrestance was obtained on a rainy day.

From Table 1 it can be seen that the 2 most serious deficiencies of the chemical papers are low efficiency on atmospheric dust and poorer obedience of the Lambert-Beer law. However, the glass fiber and millipore filters have higher pressure drops,

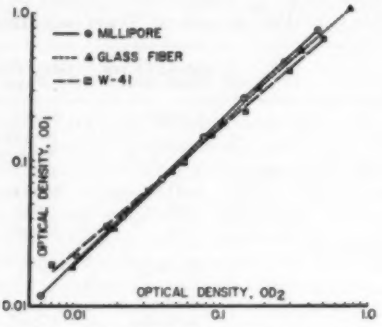


FIG. 1—DEVIATIONS FROM LAMBERT-BEER LAW FOR 3 SAMPLING MEDIA

are more fragile and require sensitive photometers because of their lower light transmission.

OPTICAL DENSITY CONCENTRATION RELATIONSHIP

Because of its general application in the field of colorimetry, the Lambert-Beer law has generally been used to derive the *OD-C* relationship[6, 7, 8]. In the usual form it is given by:

$$OD = \log (I_0/I) = kC \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Davies and Aylward[8] and Hemeon[6] have confirmed the approximate validity of Equation 2. However, more detailed investigations have shown that k is not exactly constant with C.

The decrease in k with increasing C may be evaluated directly by sampling a dust cloud of constant and known concentration[8] or by sampling simultaneously at varying rates from an unknown dust cloud. The possible influence of varying velocities through the filter medium in the latter procedure can be eliminated by using dilution with clean air to obtain different concentrations at a constant face velocity.

Because of the experimental difficulties a simpler method was devised to evaluate the relative linearity of the *OD-C* relationship for the different filter media.

Air-borne dust was sampled through 2 identical samplers at different rates. Periodically the 2 pieces of filter papers were removed and the optical density meas-

ured. For a given ratio of sampling rates through the 2 samplers, the change in the ratio of the optical densities of the filters as they load is an indication of the deviations from the Lambert-Beer law and also an indication of the error to be expected when the *OD* ratio is used as a measure of the *C* ratio in Equation 1. A convenient method of analyzing these data is to plot them on log-log paper as in Fig. 1. For such a plot

$$OD_1 = a\, OD_2^b \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

TABLE 2—OPTICAL DATA ON FILTER MEDIA

SOURCE OF DATA	DUST	MEDIA	AIR FLOW RATIO	$1 - b$		OD_1/OD_2		
				OD = 0.03 TO 0.3	OD = 0.3 TO 1	OD_1 = 0.01	0.1	1.0
Davies and Aylward[8]	Coal➤	Chemical Paper			0.14			
Engebretson[9]	Atmospheric	MF_1–MF_2	~2[a]	0.01	0.10	1.82	1.76	
Engebretson[9]	Atmospheric	W-41–MF_2	~2[a]	0.18	0.18	2.25	1.53	1.0
Present work	Atmospheric	GF_1–MF_2	1.85	0.005	0.09	2.26	2.25	
Present work	Atmospheric	MF_1–MF_2	1.93	0.005	0.06	1.98	1.96	
Present work	Atmospheric	GF_1–GF_2	1.93	0.01	0.11	1.88	1.82	
Present work	Atmospheric	W-41$_1$–W41$_2$	1.90	0.13	0.13	2.32	1.72	
Present work	Carbon dust	MF_1–MF_2	1.79	0.03	0.03	2.15	2.02	
Present work	Carbon dust	GF_1–GF_2	1.86	0.01	0.01	1.75	1.71	
Present work	Carbon dust	W-41$_1$–W41$_2$	1.91	0.06	0.06	2.0	1.61	
Theoretical-log normal distribution of A_i and k_i (σ_k = 2)			2	0.035	0.10	2.00	1.89	1.60

[a] Approximately.

where

a = the value of OD_1, at $OD_2 = 1$.

b = the slope.

If the Lambert-Beer law is obeyed exactly, the slope, b, is equal to 1. For a given air-flow ratio between filters 1 and 2, $1 - b$ may be used as an index of the deviation from the Lambert-Beer law.

Results of the authors' experiments and some similar data from Engebretson[9] and Davies and Aylward[8] are given in Table 2. From Fig. 1 it will be noted that the slope of the plot changes at an OD_1 of approximately 0.3 for the millipore and glass fiber papers. Therefore $1 - b$ in Table 2 is tabulated separately from the OD_1 ranges 0.03 to 0.3 and 0.3 to 1. The carbon dust is a semi-precipated activated carbon ground to a very fine particle size. It was dispersed with a conventional aspirator type feeder into a 12-in. square filter test duct. Isokinetic sampling was used.

From Fig. 1 and Table 2 it can be seen that the glass fiber and millipore filters well obey the Lambert-Beer law up to an *OD* of about 0.3 for atmospheric dusts and to an *OD* of about 1 for the carbon dust. However, the W-41 paper obeys the Lambert-Beer law poorly even at low optical densities.

The different behavior of the media on the carbon and atmospheric dusts indicates that the decrease in k with increasing *OD* is connected with the distribution of dust in the filter. This conclusion is further supported by the data of Table 3.

To obtain these data, equivalent amounts of dust were collected on the 3 media. It will be noted that the carbon dust which was observed not to penetrate very deeply into the glass fiber paper gave the same *OD* on millipore and glass fiber, but that the much finer atmospheric dust which somewhat penetrates the glass

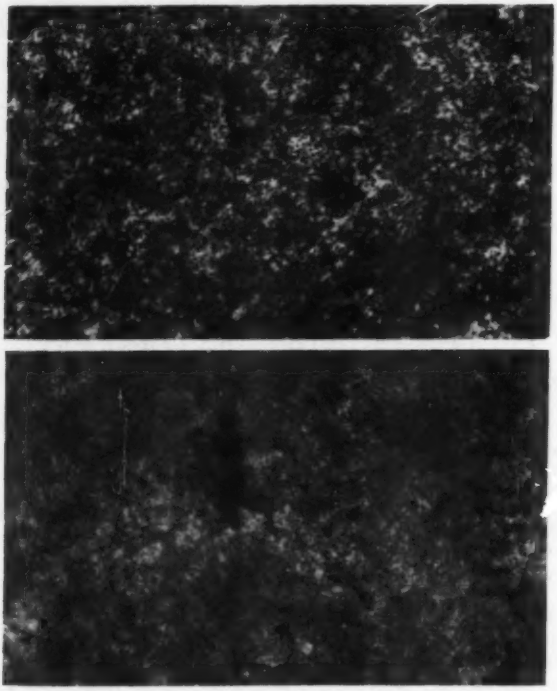


FIG. 2—VARIATION IN OD OVER SMALL AREAS. MAGNIFICATION 3.6 x (LOWER) LOADED GLASS FIBER PAPER, AND (UPPER) LOADED W-41 PAPER

fiber paper gave a higher *OD*. Apparently the greater the depth of penetration, the greater the *OD* that results from a given amount of dust.

After the much greater non-linearity of the *OD-C* relation for the W-41 paper was discovered, the characteristics of the 3 media were studied carefully to find a possible explanation. It was then noted that the small scale distribution of dust in the W-41 paper was considerably more uneven than for the millipore or glass fiber. This is shown in Fig. 2. The negatives for these photographs were made by placing the filter media in a photographic enlarger, dirty side down, and projecting the

resulting image on the film. Though the photographs are not faithful reproductions of the original degree of filter contrast the considerably poorer uniformity of the loaded W-41 paper (Fig. 2, upper) is clearly visible.

Observations on a large number of dust spot samples indicate that the larger streaks and spots sometimes observed on loaded glass fiber or millipore filters have no appreciable effect if a substantial fraction of the dust spot is included in the illuminated area of the photometer.

Further insight into the reasons for the non-linearity of the *OD-C* relationship can be obtained if the Lambert-Beer law is modified to take into account the non-uniform buildup of *OD* on the different areas of the filter.

TABLE 3—RELATIVE OPTICAL DENSITIES FOR A GIVEN AMOUNT OF DUST

MEDIA	DUST	*OD*-MEDIA *OD*-MF	COLLECTED FROM
GF	Carbon dust	1.00	Air
W-41	Carbon dust	1.14	Air
GF	Atmospheric	1.2	Air
W-41	Atmospheric	1.3	Air

If the filter has n different areas building up OD at n different rates and it is assumed that each area obeys the Lambert-Beer law and has the same initial light transmission, then:

$$I_i = I_o e^{-k_i C} \qquad (4)$$

where

I_i = the intensity of the light transmitted by the i^{th} area A_i.
C = the concentration of dust per unit area of filter.
k_i = the rate of collection of the i^{th} area.
A_i = i^{th} area.
A = total area of filter.

Then the total amount of light through the filter is IA and

$$IA = \sum_{0}^{n} I_i A_i \qquad (5)$$

and

$$I = I_o \frac{\sum_{1}^{n} A_i e^{-k_i C}}{A} \qquad (6)$$

or

$$OD = \log_{10} \frac{\sum_{1}^{n} A_i e^{-k_i C}}{A} \qquad (7)$$

That a mechanism such as that just proposed can give rise to the characteristics observed in Fig. 1 and Table 2 is shown in Fig. 3. To calculate these curves, the

approximate log-normal distribution of A_i/A and k_i given in the figure was used. Both the OD as a function of C and OD_1 *vs.* OD_2 for $C_1/C_2 = 2$ are plotted. Note that the calculated OD_1-OD_2 curve in Fig. 3 and Table 2 has a slope and curvature above $OD_1 = 0.3$ which is very similar to that observed in Fig. 1 for the glass fiber and millipore filters. Calculation of a number of such theoretical curves showed that all of the characteristics of the measured curves can be duplicated by proper choice of the distributions of k_i and A_i in Equation 7. From Figs. 1 and 3 and Tables 2 and 3 a number of conclusions may be drawn.

1. The non-linearity of the *OD-C* relationship appears to be due primarily to different rates of increase of optical density on different small scale areas of the filter.

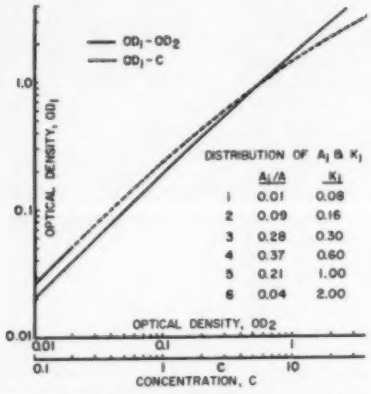


FIG. 3—THEORETICAL OD-C AND OD_1-OD_2 CURVE CALCULATED FROM EQUATION 6

2. The *OD-C* relationship is quite linear for glass fiber and millipore filters up to an OD of approximately 0.3 when sampling atmospheric dusts and up to $OD = 1$ for coarser dusts. The W-41 paper is non-linear on atmospheric dust at all optical densities and somewhat better on coarse dusts.

3. The depth of penetration determines the relative light hiding power of the dust. Millipore filters which retain the dust near the surface give lower optical densities than glass fiber or chemical papers. However the latter papers give lower optical densities when collecting coarser dusts which are retained near the surface.

4. If the ratio of downstream to upstream OD is used as a measure of the concentration ratio in Equation 1, a correction factor should be determined experimentally and used. Based on the data of Table 2 correction factors for glass fiber and millipore media have been calculated and tabulated in Table 4. The correction factor, m, is defined from Equation 8.

$$C_1/C_2 = m\,(OD_1/OD_2) \quad . \quad . \quad .

TABLE 4—APPROXIMATE OD RATIO CORRECTION FACTOR, m, FOR MILLIPORE AND GLASS FIBER MEDIA FILTERING ATMOSPHERIC DUST (Θ = 26 DEG)

OD_1	OD_1/OD_2			
	1	2	4	∞
0.02	1	1	1	1
0.05	1	1.005	1.01	1.01
0.1	1	1.01	1.02	1.03
0.2	1	1.02	1.03	1.04
0.4	1	1.04	1.07	1.08
0.8	1	1.08	1.16	1.21

Other Factors: In a previous paper[7] it was shown that from the measured size distributions of atmospheric dust, optical theory predicted the approximately correct *OD*. Since that time additional comparisons of the calculated and measured optical densities for a number of dusts have been made. Results are given in Table 5.

Measured optical densities were obtained by filtering a known amount of a benzene suspension of the dust through the filter media and by measurement of a known concentration in a Dunn cell. Optical densities were calculated according to the procedures previously described[7] except that a light scattering coefficient $K = 1$ was used. As a result, the calculated values are a little low because K is usually between 1 and 2.

From Table 5, it is readily seen that the measured values of the *OD* for the transparent dusts, Arizona road dust, glass beads is only about 1/10 of the calculated value whereas the measured *OD* of the opaque coke dust is close to the calculated value. Thus the dust spot method is not sensitive to the transparent particles in the dust. The NBS-Cottrell dust which consists of a mixture of opaque and transparent particles shows behavior in between that of the opaque and transparent dusts.

Measurements in the cell were made as a check on the other measurements. Optical densities of particles suspended in a transparent medium are relatively independent of the opacity of the particles themselves when measured at low angles of acceptance. Therefore the ratio of measured to calculated *OD* is on the order of magnitude of 1 for all dusts.

Effect of Light Wave Length on Measured OD: To discover whether there might be any advantages in measuring the optical density at a particular wave length, the

TABLE 5—RATIO OF MEASURED TO CALCULATED OPTICAL DENSITIES FOR A NUMBER OF DUSTS (K = 1)

MEASURED ON	Θ	DUST			
		ARIZONA FINE	COKE DUST	NBS-COTTRELL	GLASS BEADS
HAMF	24	0.098	1.35	0.54	0.091
GF	24	0.18	2.11	0.84	
Dunn Cell	3.3	0.57	1.89	0.62	

OD of several dusts deposited on millipore filters was measured using broad band gelatin filters. Results are given in Table 6.

Atmospheric samples 1 and 2 were taken during and just after a dust storm in which soil from the Dakotas was blown over Minneapolis. Sample No. 1 during the storm exhibits a lower ratio than the one after because it had more transparent particles than the normal city generated dust.

The large increase in *OD* near the blue end of the spectrum is rather surprising. Even if a *K* of 2 is assumed the ratios are so large for the coke and atmospheric dusts that it must be assumed that there is some additional phenomenon of interaction between particles and filters. No adequate explanation for this phenomenon has been found.

From these experiments it was concluded that there is no practical advantage in using a particular wave length band with the present photometer. It was also

TABLE 6—RATIO OF MEASURED TO CALCULATED OPTICAL DENSITIES FOR DIFFERENT LIGHT WAVE LENGTHS. (K = 1) MF FILTERS USED

GELATIN FILTER NO.	COLOR	λ mμ	DUST			
			ARIZONA FINE	COKE	ATMO. 1[a]	ATMO. 2[a]
None	White	370–700[b]	0.095	1.24	1.05	2.20
29	Red	over 605	0.082	1.37	0.97	2.23
15	Yellow	over 510	0.082	1.24	0.89	2.16
58	Green	480–600	0.18	1.50	1.24	2.57
15 + 58	Green	510–600	0.28	1.91	1.42	3.15
45	Blue	440–540	0.41	2.43	1.98	4.11
47	Blue-Violet	370–520	0.72	6.17	4.94	9.81

[a] The atmospheric samples were collected from air and evaluated by the procedures described in Ref. 7.
[b] Figures in this column are wave lengths in millimicrons.

concluded that the possible effects of different light sources and lamp color temperatures would be negligible.

IMPROVED PHOTOMETER FOR DUST SPOT LINT SCREEN AND DUST FALL MEASUREMENT

To use the glass fiber and millipore filters for dust spot measurement requires a photometer of higher sensitivity than can be conveniently obtained with the previously used simple circuits. In the simpler photometers, the output of a single self generating photocell is measured by a microammeter[5, 6, 7, 8] or in some cases is balanced against a battery as in the Wood circuit[1, 10]. These simple circuits suffer from several disadvantages. Satisfactory stability requires the use of a battery or high grade voltage regulator for the lamp power source. Also the best readibility is on the order of 0.3 percent. For these and other reasons, a much more stable balanced photocell photometer was developed.

The basic design is that described by Brice[11] and illustrated in Fig. 4. A collimated beam from the light source is split into 2 parts. One part is reflected by a first surface mirror onto the dust spot to be measured, and the remainder passes on to a reference photocell. Drift caused by unequal heating of the photocells has

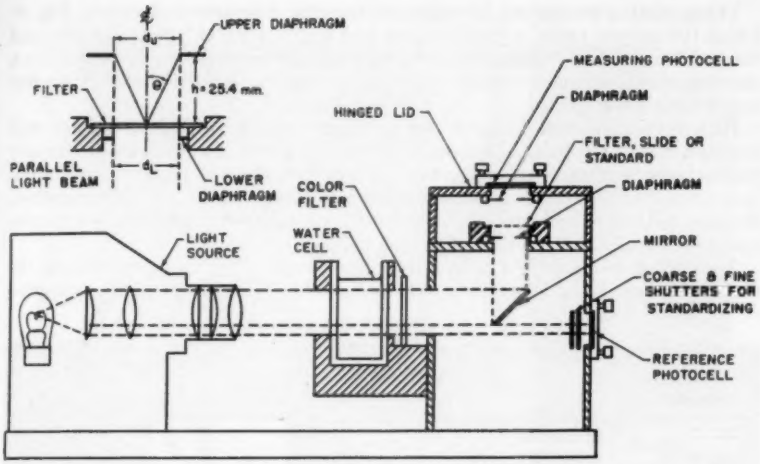
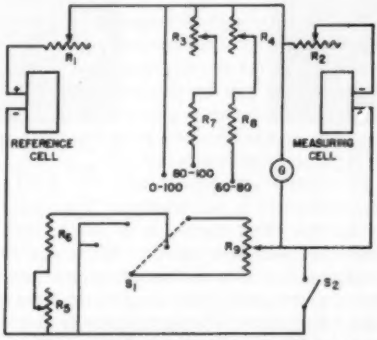


FIG. 4—OPTICAL DESIGN OF PHOTOMETER

been minimized by a water cell and placement of the light source away from the cells. Reference and measuring photocells are connected in a series circuit as shown in Fig. 5. R_9 is a 25 ohm, 10 turn potentiometer which serves to measure the ratio of the light falling on the measuring photocell to that falling on the reference cell. G is an 0.001 μA/mm sensitivity electronic null indicator.



R_1 and R_2 = 2000Ω; R_3, R_4 and R_8 = 2Ω; R_6 = 24Ω 1 percent; R_7 = 99Ω 1 percent; R_9 = 74Ω 1 percent; R_9 = 25Ω 0.1 percent linearity 10 turn pot; S-2 = DPST lever switch; S-1 = 4 pole, 3 position switch; G = 0.001 μ A/mm electronic null indicator.

FIG. 5—PHOTOMETER CIRCUIT

Three reading ranges are incorporated into the instrument shown in Fig. 6. The 0-100 percent range is used for dust spot and lint screen* measurements and the 80-100 and 60-80 percent ranges for dust fall* measurements. Switch S-2 is a shorting switch, provided to make it unnecessary to return R_9 to 100 percent when standardizing the circuit.

High output selenium photocells are to be preferred although a pair of any well matched cells can be used. Rheostats R_1 and R_2 are used to compensate for any residual mismatching of the photocells. These rheostats are adjusted until variation of the lamp voltage produces minimum movement of the galvanometer. Because of the low resistance of this circuit, low resistance switches, heavy wire and good contact must be used throughout.

An adjustable diaphragm is placed just below the filter holder to control the size of the illuminated area on the filter and another just in front of the measuring

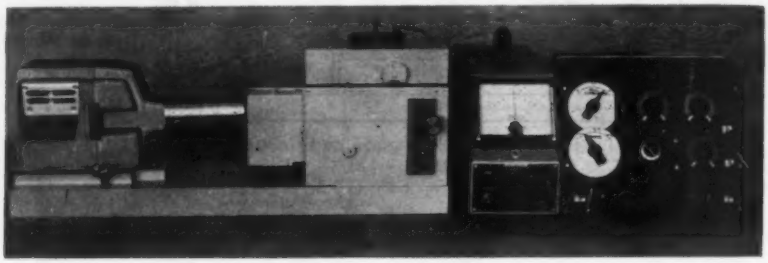


FIG. 6—IMPROVED PHOTOMETER

photocell to control its angle of acceptance. The holder shown in Figs. 7 *a*, *b*, *c*, is designed to accomodate unmounted 47 mm disks of filter media, 25 mm wide microscope slides and 2 x 2 in. glass slide covers. The hinged lid mounting the measuring photocell is held shut by a small magnetic cabinet latch.

Two shutters are provided in front of the reference photocell. The coarse shutter which slides in a groove in the end of the box is made of 2 pieces of sheet aluminum so that a piece of filter medium can be slipped between. This provides better balance between the 2 cells for dust spot measurements. The fine shutter slides in grooves in a metal mount and is adjusted by a fine pitch screw from the front. Both shutters must be rigid and free from play.

Several different light sources have been used. For most purposes a 6 volt, 20 cp lamp with suitable condensers is satisfactory. For measurements at reduced aperture on relatively opaque filter materials or when colored filters were used, a 200-w slide projector was used as a source. With any source, the light path from lamp to the photometer box must be completely enclosed to prevent instability caused by turbulent convection currents around the lenses.

Dust Spot Measurement Procedure: The instrument is first standardized by placing the appropriate standard on the holder and then adjusting the shutters with the potentiometer set at 100 or S-2 closed. The clean filter is then put in place and

* The optical lint screen and dust fall measurement techniques being used in the ASHAE dust survey will be described in a future paper.

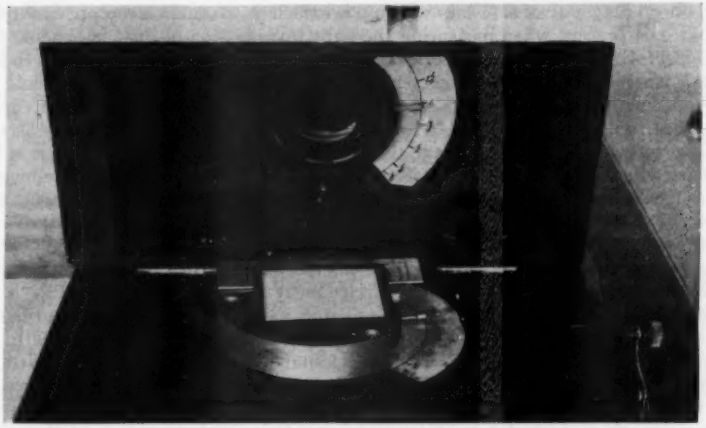


FIG. 7a—SAMPLE HOLDER OF PHOTOMETER WITH STANDARD IN PLACE

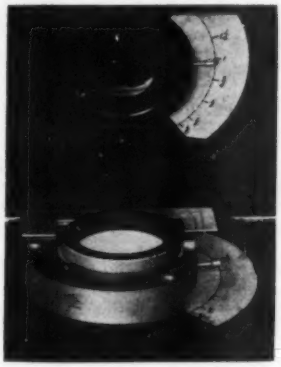


FIG. 7b—SAMPLE HOLDER OF PHOTOMETER WITH FILTER IN PLACE

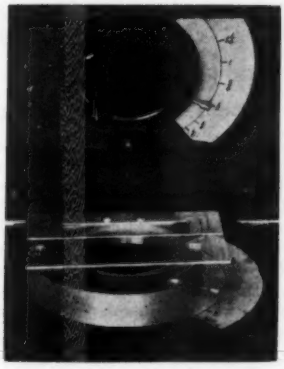


FIG. 7c—SAMPLE HOLDER WITH MICROSCOPE SLIDE

its light transmission read by turning the potentiometer until null is reached. A similar procedure is followed after the sample has been taken.

Photometer Performance: This photometer possesses a number of desirable characteristics. Readibility is excellent. On the 0-100 scale, 1 potentiometer division represents 0.1 percent transmission while on the 80-100 and 60–80 ranges, 1 division represents 0.02 percent. The rugged electronic galvanometer with its logarithmic scale makes for easy, rapid balancing with no danger of over-ranging. Sensitivity and stability are such that 0.1 percent reproducibility is easily obtained when measuring the light transmission of glass fiber or millipore filters. Stability on the 80–100

and 60–80 ranges after a 15-min warmup is sufficient to obtain a reproducibility of about 0.04 percent when measuring the light transmission of dust fall slides.

Optical Arrangement of the Measuring Photocell: In a previous paper[7] it was shown that with the particular photometer used, the *OD* of a given dust spot could be different with the dirty surface away or toward the light source. Similar effects have been noted by Davies and Aylward[8]. In a discussion of Reference 7 by Davies he suggests that this effect is due to the difference in effective scattering angle of the particles in the 2 positions. Subsequent research with the improved photometer has suggested that it is the very high angle rays leaving the filter which causes this effect.

It was found that if the light rays leaving the filter medium at angles near $\Theta = 90$ deg were prevented from reaching the photocell, then there was no measurable difference in the *OD* with the dirty surface away or toward the photocell. Most simple photometers used by other investigators as well as the one used previously[7] have the photocell mounted close to the media in a polished metal sleeve to increase photocell output. This has been found to be undesirable. Detailed investigation by Engebretson[9] has shown that if the semi-angle of acceptance, Θ, of the measuring photocell is between approximately 20 and 30 deg, then the *OD* is nearly independent of Θ. This optimum angle is obtained with $d_2 = d_1 = 25.4$ mm and $\Theta = 26.5$ deg, Fig. 4. The diaphragms and the inside of the photometer lid are flat black to prevent high angle rays from reaching the photocells.

A further advantage of this optical arrangement is that the Lambert-Beer law is obeyed better at higher optical densities.

CONCLUSIONS

From this study of the dust spot method, a number of conclusions have been reached.

1. For precision dust spot measurement of air cleaner performance, glass fiber paper and millipore filters have been found superior to the chemical papers ordinarily used.
2. A study of the *OD-C* relationship for the different media has shown that the glass fiber and millipore filters are very nearly linear for optical densities below 0.3, while the W-41 chemical paper is relatively non-linear at all optical densities.

The linearity of the *OD-C* relationship has also been found to depend on the optical system of the photometer and on the dust being sampled.

3. A theoretical explanation for the non-linearity of the *OD-C* relationship has been proposed. This theory indicates that the non-linearity is due to unequal rate of *OD* increase on different areas of the sampling media.
4. The optimum semi-angle of acceptance of the measuring photocell has been found to be between 20 and 30 deg. An optimum optical design has been recommended.
5. An improved null type photometer having greater sensitivity, accuracy, convenience, and flexibility has been developed for dust spot measurement. The greater sensitivity of this photometer is necessary to utilize the relatively opaque glass fiber and millipore filters under the optimum conditions of measurement recommended.

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NOMENCLATURE

A = filter area.
C = dust loading per unit filter area.
q = sampling rate.
t = sampling time.
OD = optical density.
I_0 = light transmission of clean filter.
I = light transmission of dirty filter.
k = constant.
p = pressure drop across filter media. $p = p_0 V^n$
p_0 = resistance coefficient, in centimeters of mercury.
V = filter face velocity, in feet per minute.
n = a constant.
K = light scattering coefficient.
η = filter arrestance or efficiency.
Θ = semi-angle of acceptance of measuring photocell.
u = subscript referring to upstream.
d = subscript referring to downstream.

DISCUSSION

W. C. L. HEMEON, Pittsburgh, Pa. (WRITTEN): The authors have presented some useful data on this subject, the validity of which cannot be questioned. One may, however, debate the significance of particular data and we should like therefore to question some of the conclusions that have been advanced. An examination of graphs of Fig. 1 certainly show some deviation from the Lambert-Beer law for paper W-41 but the magnitude of the deviation, as we see it, is insignificantly small in considering its application to evaluation of air cleaners. The practical significance could be demonstrated in quantitative terms by plotting optical density against air quantity, drawing the line of best fit through the origin and then considering the size of the error for those points that stand away from the line. The original data for such calculations are not presented in the paper, although it is easy to see how, in Fig. 1, a line of slope = 1 could readily be drawn through the W-41 points without severe distortion. In sum, we believe that in the authors' discussion the phrases . . . *W-41 paper obeys the Lambert-Beer law poorly* . . . and reference to *the much greater non-linearity . . . for the W-41 paper* tend to exaggerate the significance of the deviation in relation to practical consequences. It would be appreciated if the authors would present some conclusions as to actual

percentage error in the use of W-41 filter paper based on the method of data treatment we have described.

R. D. RIVERS‡, Louisville, Ky.: I find myself in the middle between Dr. Whitby and Dr. Hemeon, in that I do believe that the deviations from Lambert-Beer Law are important in this matter but I do not feel that the Lambert-Beer Law has any particular sacredness. If the percent transmission-dust load relationship can be properly described—which it can—for any one of these media, then we can properly correct for the deviations from linearity.

Our experience has been that while certainly the microscopic deviations of build-up of dust on glass fiber and millipore are greater with W-41, the macroscopic deviations are not. That is, if you set up 3 samplers drawing equal quantities of air through equal diameter targets and measure the percentage of change in the light transmission of these targets after exposure to the atmosphere, the glass fiber paper I am familiar with, (and I don't know whether I used the same ones that Dr. Whitby used) showed the greatest variation, the millipore the next, and the W-41 the least variation. This we can only ascribe to large variations in the densities of millipore and glass fiber paper which are quite visible to the naked eye. There are weak points where the dust tends to collect and strong points where it does not. Our experience along this line would seem to be somewhat at variance from the experience in the paper.

AUTHORS' CLOSURE (Dr. Whitby): The significance of the non-linearity of the relationship between optical density and concentration mentioned by Mr. Hemeon, will depend on the application. This non-linearity will not be important for the measurement of airborne dust stain concentration with such instruments as the *AISI* sampler. It will be important, however, when measuring the dust spot arrestance of air cleaners.

The factor m tabulated in Table 4 is a direct measure of the error to be expected. Note that for an OD_1/OD_2 ratio of 4 that an error of 16 percent could result if OD_1 equaled 0.8.

The magnitude of the possible error will depend on such things as the kind of dust, sampling media, photometer optical system and the depth of penetration into the media. For some test dusts and apparatus designs, it is possible that 30 percent errors in calculating the concentration ratio could result from using the uncorrected *OD* ratio in place of the concentration ratio.

I agree with Mr. Rivers that the Lambert-Beer law should not be considered sacred. It does serve however as the best base for any system of correction or for the development of improved theories.

Our experience with the reproducibility of the W-41, glass fiber and millipore is the opposite of that of Mr. Rivers. In our original work the accuracy of our sampling system was such that a reproducibility of about 3 percent standard deviation was obtained. These experiments have been repeated recently with a much improved system in which a standard deviation of about 0.5 percent was obtained with the glass fiber and about 1 percent with the millipore. During these experiments it was found that practically all of the error could be traced to errors in air flow measurement, sampler clamping arrangement or photometer measurement error. It is possible that the greater error with the glass paper observed by Mr. Rivers is due to the samplers and air flow measurement system used, because the clamping problems and pressure drop of the 3 media are quite different.

Since the writing of the paper we have also learned that glass fiber paper without binder must be used for maximum reproducibility.

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‡ American Air Filter Co.

1598

No. 1598

COLD WALL EFFECTS IN A CEILING-PANEL-HEATED ROOM

By L. F. SCHUTRUM* AND T. C. MIN**, CLEVELAND, OHIO

This paper is the result of research carried on by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio.

THIS REPORT concerns the combined effect of a cold wall or window and a heated ceiling panel on room-air temperature, mean radiant temperature (MRT) and air motion. All 3 of these factors are known to influence human comfort, but at present there is no accepted method of evaluating their combined effect.

Tests were made with heated panel areas near the cold wall to determine the extent to which such an arrangement could increase the air temperature and the radiant temperatures near the cold surface.

A project will be undertaken at the Research Laboratory of the Society in the near future to determine the effects of radiation and other environmental factors on human comfort. When these studies are completed, the data presented in this paper can be evaluated in terms of comfort effects.

HOW INVESTIGATION WAS MADE

Late in 1955, the Society undertook studies of the cold wall effect as an extension of the long-range research program which has been guided since its inception in 1947 by the Technical Advisory Committee on Panel Heating and Cooling†.

The ASHAE Environment Room[1] in which the tests were made is 24½ ft long, 12 ft wide, and 9½ ft high (7.5 x 3.7 x 2.9 m). The room is described and illustrated

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† P. B. Gordon, chairman; W. S. Harris, vice chairman; A. B. Algren, W. P. Chapman, John Everetts, Jr., H. T. Gilkey, E. S. Howarth, L. N. Hunter, A. T. Jorn, W. C. Kadow, G. D. Lain, H. A. Lockhart, C. O. Mackey, R. L. Maher, S. K. Smith E. F. Snyder, Jr., W. F. Spiegel, J. M. van Nieukerken, G. L. Wiggs.

[1] Exponent numerals refer to References.

Presented at the 63rd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Chicago, February 1957.

in Reference 1. All surfaces of the room are composed of aluminum panels through which liquid at any desired temperature can be circulated. Surface temperatures are measured by about 400 copper-constantan thermocouples, and heat flow through the surfaces is determined by plate-type heat-flow meters[3].

Measurements of room-air temperatures and velocities, room-surface temperatures, radiation, and heat flow were made under steady-state room conditions of fixed ventilation, one cold sidewall, a heated ceiling panel of various widths, all other surfaces being neither heated nor cooled by liquid circulation.

In each test the entire area of one long wall of the room was controlled at a selected temperature, while the temperature of the heated section of the ceiling was

FIG. 1—VIEW OF GLASS WALL IN TEST ROOM

adjusted to maintain a room-air temperature of 70 F (21 C). The temperatures of the cold wall were selected arbitrarily to cover the range of room-surface temperatures found in practice. The ceiling panel was adjacent to and the full length of the cold wall, and was 4 ft, 8 ft, or 12 ft in width (1.2, 2.4, and 3.7 m). The latter constituted the whole width of the ceiling. The floor, the other 3 walls, and the unheated portions of the ceiling were valved-off from the circulating temperature-controlling fluid and thus reached an equilibrium temperature resulting from the radiation and convection exchanges in the room and the heat exchange at the back of the panels. The temperature of the air behind the panels was controlled at approximately 70 F (21 C), and because the backs of all panels were insulated, the uncontrolled walls, floor, and sections of the ceiling were in effect, thermally neutral.

Ventilation air for 1.5 room-air changes an hour (0.25 cfm per sq ft of floor area) at 70 F was introduced into the room through two 8-in. (20.8 cm) ceiling diffusers

located in the center section of each half of the ceiling. The throw of air was directed horizontally from the diffusers which protruded 2 in. from the ceiling. The air left the room through a long-narrow slot located between the floor and the wall opposite the cold wall. The dew-point temperature of the room air was maintained below the temperature of the cold wall to prevent condensation. The temperature of the cold wall was essentially uniform.

To determine the temperature variation which would occur on a wall not directly controlled by fluid circulation, one test was made with one-half of the cold wall covered with an inner wall of double-strength glass spaced about 4 in. away from the cold wall and extending from floor to ceiling. This glass wall may be seen in Fig. 1.

Room-air temperatures were sensed by 36-gage copper-constantan butt-welded thermocouples which were found not to be appreciably affected by radiation. Room-air velocities were measured with a commercial heated-wire type anemometer and verified by a heated-thermocouple anemometer[3], both of which were calibrated prior to use.

Radiation was measured by means of globe thermometers[4, 5] and a flat 2π radiometer[6] which was calibrated before the tests were begun. The 3 globe thermometers which were used were 6 in. in diameter and were coated with a mixture of lamp black and shellac having a normal emissivity of 0.95[7]. The equilibrium globe temperatures result from a balance between radiation gain and the convection loss.

The 2π radiometer was set up at several locations in the room, and in each location, a reading was taken normal to each of the 6 room surfaces. From the directional radiant temperatures thus obtained the mean radiant temperatures were calculated.

An indicating electronic potentiometer having an emf scale and a special direct-reading temperature scale was used to measure the electrical output of thermocouples, heat-flow meters, radiometers, and one of the anemometers.

The test room was unlighted and unfurnished, and was unoccupied except when air temperature, globe thermometer, radiometer and anemometer readings were being taken. During these readings it was necessary for one person working by flashlight, to be in the room to position the instruments.

TEST RESULTS

Table 1 presents a summary of room-surface temperatures, air temperatures and panel areas for all of the tests. The air temperatures given in this table were taken with the room unoccupied. All other air temperatures given in the papers for the same location were taken with one occupant in the room, and are slightly higher because of this occupancy.

The highest panel temperatures and the lowest cold-wall temperatures given were used for Laboratory purposes only to establish the trends of the factors, and should not necessarily be regarded as good practice for producing comfort conditions.

ROOM AIR TEMPERATURES

The distribution of the room-air temperatures in a plane through the center of the room is shown in Figs. 2, 3, and 4 for three combinations of cold-wall and panel

temperatures. Air-temperature measurements were also made in planes 6 ft and 1 ft away from an end wall, and were found to be essentially the same as those shown in the figures.

Note the uniformity of air temperature from center of room to within 6 in. of the cold wall. Fixed thermocouples which could be read nearly simultaneously indicated no horizontal gradient at the 60-in. level from the center of the room to within 2 in. of the cold wall. In Fig. 2, with the extremely cold wall, the air at the 6- and 12-in. levels is warmer near the cold wall than in the center of the room; however, at the 2-in. level, the opposite is true. This is probably due to the thin

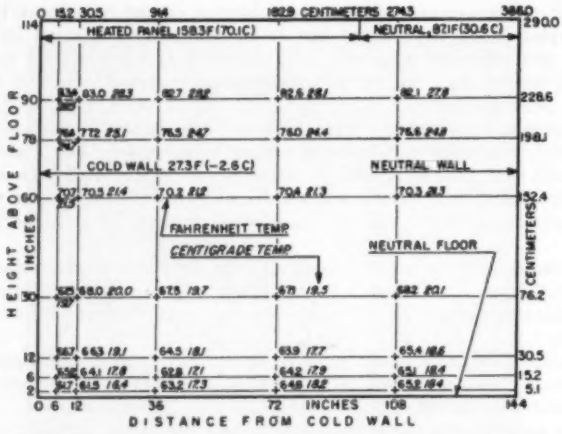

FIG. 2—ROOM-AIR TEMPERATURE DISTRIBUTION, CENTER OF ROOM, COLD WALL 27.3 F (−2.6 C), TEST NO. 510

layer of cool air flowing down the wall and across the floor in widening layers. The air-flow pattern will be discussed later.

The results given in Figs. 2, 3, and 4 were obtained with an 8-ft wide heated panel. Tests made with wider and narrower panels indicated that the width of the panel did not appreciably alter the air-temperature distribution in the living space *i.e.*, up to 6½ ft (2.0 m) above the floor.

The floor-to-ceiling air-temperature gradients shown in Figs. 2, 3, and 4 were about the same as those previously reported[8] for a cold-wall condition. A comparison of the air-temperature gradients at a position 1 ft from the cold wall can be made from Fig. 5A for 3 cold-wall temperatures.

The air-temperature gradients shown in Fig. 5A were measured under conditions of a uniform temperature of the cold wall. The question arose as to the effect a non-uniform cold-wall temperature, which is the usual condition, would have upon the air temperature near the wall. The test with the inner glass wall covering one-half of the cold wall furnished the answer. In this test the surface temperature of the uncovered cold wall was adjusted to be the same (within 1 F deg) as the average surface temperature of the glass wall. The variation in temperature from top to

TABLE 1—TEST CONDITIONS

TEST NO.	COLD-WALL TEMP.		CEILING						ROOM-AIR TEMP. (5 FT HEIGHT, 1.52 M) OBSERVED		FLOOR[b] TEMPERATURE AVERAGE		WALL[b] TEMPERATURE AVERAGE	
			PANEL WIDTH		TEMPERATURE									
					HEATED[a]		UNHEATED[b]							
	F	C	FT	METER	F	C	F	C	F	C	F	C	F	C
505	59.7	15.4	4	1.22	96.4	35.7	72.4	22.5	69.2	20.7	70.2	21.2	70.4	21.3
511	60.1	15.6	8	2.44	88.1	31.1	73.8	23.2	70.1	21.1	71.2	21.8	71.3	21.8
512	60.5	15.8	12	3.66	83.4	28.6	NONE	NONE	70.2	21.2	71.0	21.7	71.6	22.0
513	54.4[c]	12.4	8	2.44	98.4	36.8	74.2	23.4	69.6	20.9	71.0	21.7	71.4	21.9
506	42.1	5.6	4	1.22	171.2	77.3	78.5	25.8	70.1	21.1	73.9	23.3	72.9	22.7
507	41.5	5.3	8	2.44	130.5	54.7	81.1	27.3	70.4	21.3	74.8	23.8	74.4	23.5
508	41.6	5.3	12	3.66	114.6	45.9	NONE	NONE	69.9	21.0	74.2	23.4	75.3	24.1
510	27.3	−2.6	8	2.44	158.3	70.1	87.1	30.6	70.0	21.1	78.3	25.7	77.7	25.4
509	27.0	−2.8	12	3.66	137.5	58.6	NONE	NONE	69.9	21.0	77.7	25.4	77.2	25.1

[a] Panel temperature required to give observed room-air temperature.
[b] Neutral temperature, surface neither heated nor cooled.
[c] Glass temperature avg 55.4 F, with cold surface behind glass at 25 F.
Ventilation air 75 cfm at 70 F for all tests.

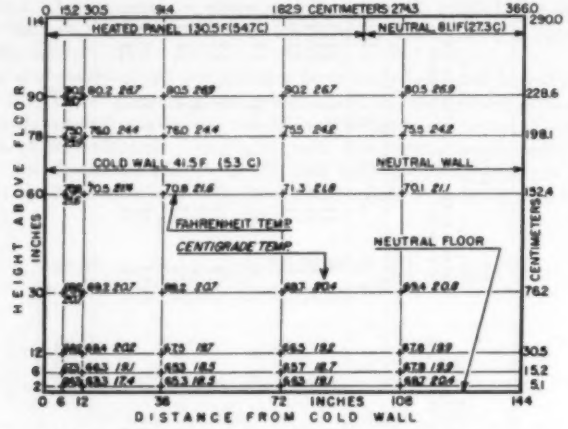
0 152 305 914 1829 CENTIMETERS 2743 3660
HEATED PANEL 130.5 F (54.7 C)
NEUTRAL 81.1 F (27.3 C)
COLD WALL 41.5 F (5.3 C)
NEUTRAL WALL
FAHRENHEIT TEMP
CENTIGRADE TEMP
NEUTRAL FLOOR
HEIGHT ABOVE FLOOR INCHES
CENTIMETERS
DISTANCE FROM COLD WALL
0 6 12 36 72 108 144 INCHES

FIG. 3—ROOM-AIR TEMPERATURE DISTRIBUTION, CENTER OF ROOM, COLD WALL 41.5 F (5.3 C), TEST NO. 507

bottom of the exposed cold wall was about 3 F (1.7 C) and for the glass wall about 15 F (8.3 C) (see section on Surface Temperatures).

Air temperatures 1 ft from the center of the exposed cold wall and the glass wall are shown in Fig. 5B. As may be seen from the figure, very little difference exists

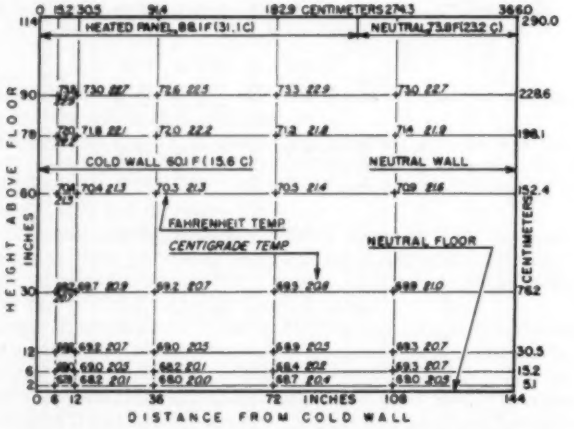
0 152 305 914 1829 CENTIMETERS 2743 3660
HEATED PANEL 88.1 F (31.1 C)
NEUTRAL 73.8 F (23.2 C)
COLD WALL 60.1 F (15.6 C)
NEUTRAL WALL
FAHRENHEIT TEMP
CENTIGRADE TEMP
NEUTRAL FLOOR
HEIGHT ABOVE FLOOR INCHES
CENTIMETERS
DISTANCE FROM COLD WALL
0 6 12 36 72 108 144 INCHES

FIG. 4—ROOM-AIR TEMPERATURE DISTRIBUTION, CENTER OF ROOM, COLD WALL 60.1 F (15.6 C), TEST NO. 511

between the air temperatures near the uniform temperature wall and the non-uniform temperature wall (glass).

AIR MOVEMENT

The direction of air motion was determined by observation of cigarette-smoke patterns. In general, the flow of air consisted mainly of a very thin air film in contact with the cold wall which fell rapidly down the wall with increasing velocity and thickness, then flowed across the floor, rising slightly, with part of this air carrying up the opposite wall. The air flow near the ceiling was slight except in the vicinity of the ceiling diffusers which introduced the ventilation air. The bulk

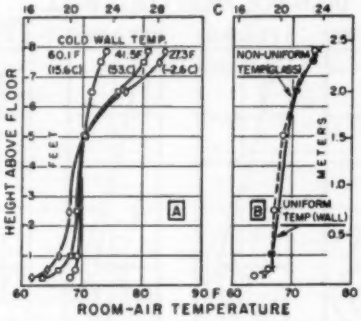

FIG. 5—ROOM-AIR TEMPERATURE GRADIENTS, 1 FT (0.3 M) FROM COLD WALL, FOR VARIOUS COLD WALL TEMPERATURES, 8 FT (2.4 M) PANEL WIDTH

of the room air was relatively still. As the air flow down the cold wall was not in the normally occupied space, only the flow across the floor was considered to be important.

The velocity of the air flow across the floor is shown in Fig. 6. The air velocities are the average of the maximum and minimum observations taken over a period of about 15 to 90 sec. The flow of air shown in the figure is away from the cold wall, decreasing in velocity and increasing in height. The velocity of the air was greater with the lower wall temperatures, but it did not change noticeably with the size of the heated ceiling panels.

The velocity of the air flowing across the floor was also measured without ventilation air being supplied to the room, for the surface-temperature conditions of the test No. 510. The measured air velocities across the floor were about the same, with or without ventilation.

No significant difference in the velocity of air flow across the floor was noted between the uniform cold-wall temperature condition and the non-uniform or glass wall. However, the velocity of the air flowing down the glass partition was slightly

less at the top and more at the bottom than for the wall of the same but uniform temperature.

FLOOR-SURFACE TEMPERATURE

The temperature of the surface of the floor is recognized as contributing to foot comfort for adults and body comfort for children playing on the floor.

Figs. 7A and 7B show the effects of the temperatures of the cold wall and heated panel and of panel width on floor-surface temperatures. In Fig. 7A, the floor temperature is shown to be highest for the highest panel temperature, despite

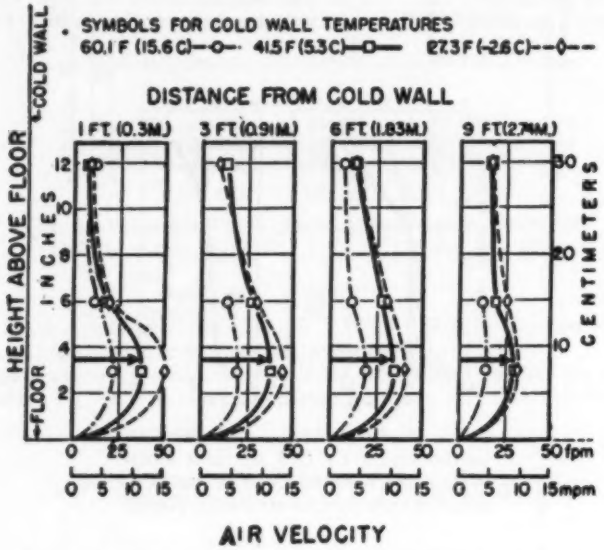

FIG. 6—AIR VELOCITY NEAR FLOOR FOR THREE COLD WALL TEMPERATURES, 8 FT (2.4 M) PANEL WIDTH

the lower cold-wall temperature. The width or location of the heated ceiling panel influences the temperature of the floor as shown in Fig. 7B. The narrow 4-ft-wide panel warms the floor surface closer to the cold wall more than do the wider 8-ft and 12-ft ceiling panels. Because the floor surface was neutral and the average floor-surface temperature was higher than the room air, the floor must have received a net gain of radiant energy from the surroundings, which was given up to the room air by convection. The warm floor, therefore, contributed to the uniformity of the room air temperature.

GLOBE THERMOMETER TEMPERATURES

The globe thermometers measure the combined effect of air temperature, air movement, and radiation. Where still-air conditions exist and humidity effects are negligible, the globe temperature is sometimes used as an environmental index[9].

Observed globe temperatures taken at the 60- and 30-in. levels at the center of the test room are shown in Fig. 8. The temperatures at the 12-in. level (not shown) were much more influenced by air movement, but near the cold wall were within one deg F of those shown in Fig. 8 for the 30-in. level. In the center of the room, however, the difference in globe temperatures at the 30- and 12-in. levels ranged up to about 3 F for the lowest temperature cold-wall condition.

In Fig. 8A the globe temperatures are higher for the colder wall conditions. This is the result of the warmer ceiling panels required to maintain a 70 F air temperature (see Table 1) and the warmer surface temperature of the neutral three

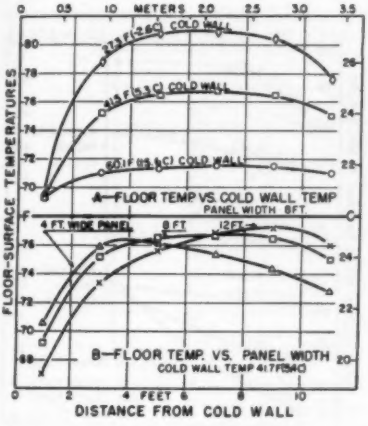

FIG. 7—FLOOR-SURFACE TEMPERATURES

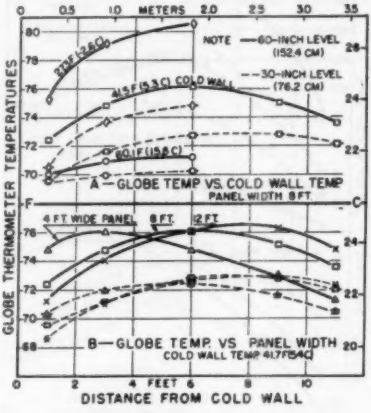

FIG. 8—GLOBE THERMOMETER TEMPERATURES

walls and the floor. In most locations the influence of radiation is sufficient to raise the globe temperature above that of the room air.

The effect on the globe temperatures of the size and also location of the heated panel is shown in Fig. 8B. In the vicinity of the cold wall, the smaller, warmer ceiling panel causes the globe temperatures to be higher, whereas near the neutral wall, opposite the cold wall, the influence of the smaller ceiling panels is less and the globe temperatures are lower.

MEAN RADIANT TEMPERATURE

The radiometer measures a temperature which is the equivalent black-body temperature of the surfaces which the radiometer sees. The radiometer surface is flat, and receives radiation from all directions of a hemisphere. The six directional readings previously described were averaged to obtain the mean radiant temperature. The result of a 6 reading survey was judged to be acceptable only after it was found to check satisfactorily with the result of a 26 reading survey. This more extensive survey included the 6 readings taken normal to the 6 room

surfaces, together with all 20 additional 45 deg angular readings which could be taken about a point.

The mean radiant temperatures are shown in Fig. 9A, as a function of position and cold-wall temperature. The mean radiant temperature is shown to be highest for the coldest wall temperature due to the higher required panel temperature. Similarity is notable between this figure, which shows the radiation effect, and of Fig. 8 (globe temperature), which integrates the radiation and convection. The mean radiant temperatures were higher than the globe temperatures. The mean

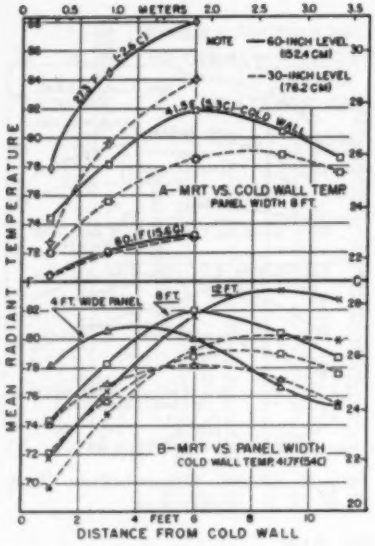

FIG. 9—MEAN RADIANT TEMPERATURE

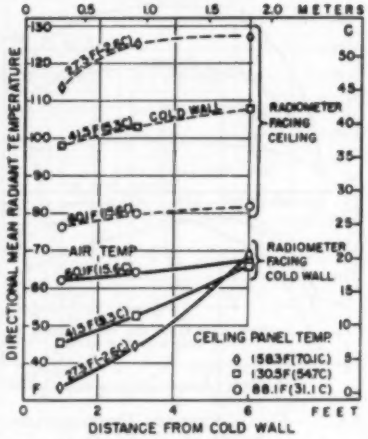

FIG. 10—DIRECTIONAL MEAN RADIANT TEMPERATURE, 60-IN. (152.4 CM) LEVEL, 8 FT (2.4 M) WIDE PANEL

radiant temperature is considered by many as an important part of a comfort index.

Fig. 9B shows the effect of the width and temperature of the heated part of the ceiling upon the mean radiant temperature. The MRT in both Figs. 9A and 9B are elevated above the 70 F room-air temperature in all of the occupied space. The difference between MRT and room-air temperature would be extreme in an unfurnished room. In a previous study[8] the room-air temperature in a furnished room was found to be higher than in an empty room having the same room-surface temperatures. Thus, room furnishings would tend to reduce the elevation of the MRT above that of the room air.

DIRECTIONAL MEAN RADIANT TEMPERATURES (DMRT)

The human body is in fact a thermal integrater. To what extent the body may integrate warmth on one side and coolness on the other has not been fully investi-

gated. Some excellent work has been done in this direction by F. A. Chrenko[10]. Whether the comfort criteria for radiation should be founded on a mean radiant or some directional radiant temperature[11] will be studied in the Society's long-range program on human comfort. However, the directional radiant temperatures reported in Figs. 10 and 11 are of interest in guiding the comfort investigation. In both figures the directional mean radiant temperatures are those obtained with the radiometer facing in the direction of the cold wall or the ceiling.

In Fig. 10, for the 8-ft-wide panel, the directional mean radiant temperature DMRT facing the cold wall approached the temperature of the cold wall as the distance from it decreased. In the direction of the ceiling, the DMRT was the highest for the test in which the panel temperature was highest and the cold-wall temperature was the lowest (27 F). Obviously, there is an upper limit of panel temperature satisfactory for comfort.

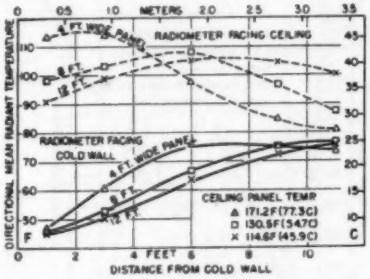

FIG. 11—DIRECTIONAL MEAN RADIANT TEMPERATURE, 60-IN. (152.4 CM) LEVEL, COLD WALL TEMPERATURE 41.7 F (5.4 C)

Fig. 11 shows the effect of panel width and temperature on the DMRT for a given cold-wall temperature. It may be noted that the DMRT in the direction of the cold wall increases as the panel width decreases and the panel temperature increases. In the direction of the ceiling, the DMRT near the cold wall also increases as the panel temperature increases, despite a decrease in panel width.

No difference was found in the DMRT as measured 1-ft away from the non-uniform glass partition and 1-ft away from the equivalent isothermal cold wall.

Both the mean radiant temperature and the directional mean radiant temperature could be calculated satisfactorily on the assumption that all surfaces were thermally black. Because of the high emissivity of the surfaces, reflections were quickly damped out. The accuracy of the calculated directional mean radiant temperatures improved, compared with observed, when the areas of the surfaces with non-uniform temperatures were divided into smaller divisions[12].

HEAT FLOW

The heat loss from the test room must be matched essentially by the heat input to the room from the heated ceiling panel. The heat output of the ceiling panel

was measured by 32 plate-type heat-flow meters which covered about 10 percent of the ceiling area.

The lower curve of Fig. 12 shows the unit panel output of the heated part of the ceiling *vs* the temperature difference between the ceiling and the room air. The observed heat flow rates are shown to lie on the curve regardless of the width of the heated panel. Included in Fig. 12 are the average heat-flow rates from the entire ceiling, plotted against the difference between the average ceiling temperature and the room-air temperature. These points follow closely the curve drawn for the heated panels only.

A comparison of the ceiling heat-flow curve of Fig. 12 with the ceiling panel outputs reported in the Society's paper on Thermal Design of Warm Water Ceiling

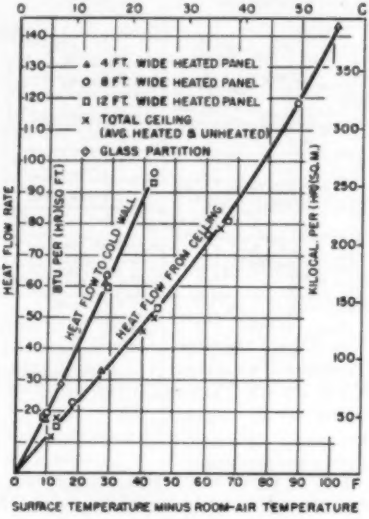
4 FT. WIDE HEATED PANEL
8 FT. WIDE HEATED PANEL
12 FT. WIDE HEATED PANEL
TOTAL CEILING (AVG. HEATED & UNHEATED)
GLASS PARTITION
HEAT FLOW RATE
BTU PER (HR)(SQ FT)
KILOCAL. PER (HR)(SQ. M)
HEAT FLOW TO COLD WALL
HEAT FLOW FROM CEILING
SURFACE TEMPERATURE MINUS ROOM-AIR TEMPERATURE

FIG. 12—HEAT-FLOW RATES FOR CEILING AND COLD WALL

Panels[13] shows general compatability of the data in the 2 studies. Therefore, the equations which were used in developing the design procedure may be applied without serious discrepancy to the heated panel in these tests. The basic equations are reported in an earlier paper[14]. The radiation equations are valid, as the surface emissivities (0.92 to 0.93) were nearly the same as those previously reported[14].

The average heat-flow rates through the cold wall are also shown in Fig. 12. Heat flows calculated from the radiation and convection equations reported in reference 14 were found to be in reasonable agreement with the observed values.

Fig. 13 shows the variation of the heat flow through the cold wall with height. Each plotted point was determined by averaging the readings of all of the heat-flow meters on the cold wall located at a given height. The 3 curves illustrating

the effect of the temperature of the cold wall (ceiling panel 8-ft wide) show that the heat flow for the colder wall is not only higher, but also the vertical gradient is greater. The 3 central curves, with a constant 41.7 F cold-wall temperature illustrate the effect of panel width and temperature upon the heat-flow gradient. The narrower but warmer 4-ft-wide panel causes greater heat flow through the upper section of the cold wall and less through the lower section than do the wider ceiling panels. Both radiation and convection contribute to the heat flow. However, it is likely that the radiation exchange is largely responsible for the variation of heat-flow gradients with the width and temperature of the heated panel.

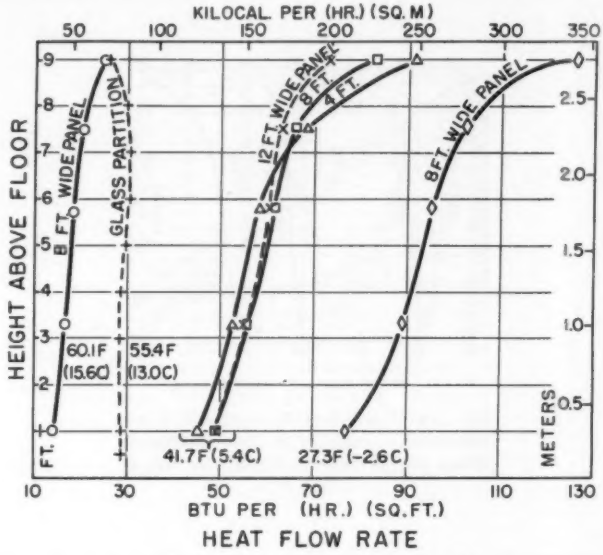

FIG. 13—HEAT FLOW INTO COLD WALL

The heat flow into the glass partition was measured by commercial-type heat-flow meters, which were pressed against the glass surface at desired locations. The absolute rates of heat flow obtained by this method were not considered precise. These heat-flow rates were corrected to give the same average rate as measured in the same test and temperature for the isothermal cold wall. The resulting heat-flow rates through the glass are shown in Fig. 13. The convection between the glass in the partition and the air in the space behind the glass is responsible for the relatively uniform heat flow. The effect that convection in the wall space has upon surface temperature and heat flow through the wall has been investigated thoroughly for an air-heating system[15,16].

SURFACE TEMPERATURES

The surface of an actual exposed wall or window is not uniform in temperature, and since in these tests the heat is supplied at the ceiling, it is particularly desirable

to know what surface-temperature gradients would exist for an actual wall under the test-room conditions. Calculation of these temperatures was attempted by the method described in the Appendix. The main purpose of the test with the glass partition was to obtain a non-uniform surface temperature against which the calculated surface temperatures could be compared. These are given at the right-hand side of Fig. 14. The calculated temperatures are greater above the 8-ft level and less below 2 ft than are the observed. On the basis of temperature difference, the convection transfer to a non-uniform temperature cold wall would be less near the top and more near the bottom of the wall than for an isothermal wall. This may be the reason for the difference between calculated and observed glass-

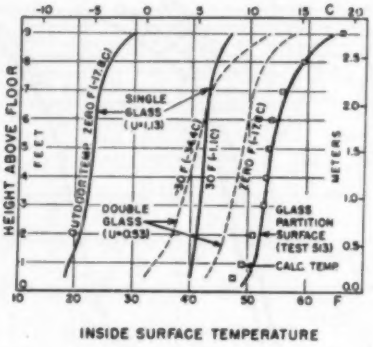

FIG. 14—CALCULATED SURFACE TEMPERATURE GRADIENTS FOR EXPOSED WALL OR WINDOW, ROOM AIR 70 F (21.1 C)

surface temperatures, because the thermal conductances used in the calculation were obtained from isothermal wall conditions.

The calculated surface-temperature gradients shown by the other curves of Fig. 14 may also be slightly greater than the true gradients and should be weighed accordingly. (See Appendix for calculations). A double-glass window is shown in Fig. 14 to have about the same average inside surface temperature with −30 F (−34.5 C) outdoors as has single-glass +30 F (−1.1 C) outdoors. The temperature gradient for the double-glass, nevertheless, is much greater. As pointed out previously this gradient may be slightly greater than the true gradient.

The average surface temperatures of single and double glass and a wall having a *U*-value of 0.17 Btu per (hr) (sq ft) (F deg) were calculated by the method described previously, and are given in Fig. 15 for various outdoor temperatures. For comparison the surface temperature is given for single glass $U = 1.13$ or 5.52 kilocal per (hr) (sq m) (C deg) with a fixed inside coefficient, surface to air, of 1.50 or 7.33 kilocal per (hr) (sq m) (C deg). This *U*-value is reported in Table 20 of THE GUIDE, 1955, for a single sheet of glass in a vertical position. The inside film coefficient, 1.5, is an average value for vertical glass and glass block calculated

on the assumption that the room-air temperature equals the average temperature of the room surfaces seen by the glass. This condition does not necessarily prevail in an actual room, and the curve based on a film coefficient of 1.5 cannot be considered representative for any type or arrangement of heating system.

The inside coefficient for test conditions may be obtained from Fig. 12 by dividing the cold-wall heat-flow rate by the temperature difference between room air and the cold wall. This averages about 2 Btu per (hr) (sq ft) (F deg) or 9.76 kilocal per (hr) (sq m) (C deg) for a temperature difference of 10 to 15 F (5 to 8 C). As pointed out previously, furniture in a room tends to increase the room-air temperature and the inside film coefficient may be somewhat lower in an actual room.

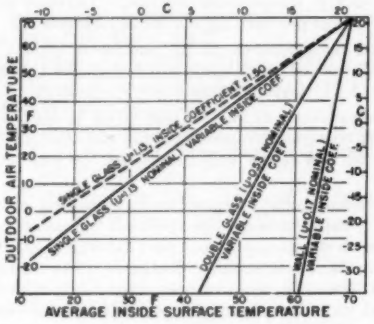

FIG. 15—AVERAGE INSIDE SURFACE TEMPERATURE OF EXPOSED WALL OR ALL GLASS WALL AS CALCULATED FOR ROOM HEATED BY CEILING PANEL, ROOM AIR 70 F (21.1 C)

Previous data also indicated that the heat flow through a cold wall, and therefore the film coefficient, may increase slightly with increase in cold-wall height.

CONCLUSIONS

On the basis of tests conducted in a room with one cold wall and heated by ceiling panels adjacent to the cold wall, the following conclusions were drawn:

1. Variations in room-air temperatures were small in the living space to within inches of the cold wall, in any horizontal plane above approximately the 6- to 12-in. level (15.2 to 30.5 cm). Below this level the air temperature was influenced by air flowing down the cold wall.

2. The vertical gradients of room-air temperature were minimum when the cold wall and heated panel were nearest 70 F. The vertical temperature gradients between the 2-in. and 60-in. levels in the room increased to only 5.6 F deg (3.1 C deg) at the extreme condition when the cold wall was 27.3 F (−2.6 C) and the 8-ft wide ceiling panel was at 159.4 F (70.8 C). Under the usual practical conditions it would be expected that the gradients would be less.

3. Air flowing down the cold wall travels across the floor in a layer about 12 in. or less in thickness. Average velocities measured within this layer were up to 40 fpm (12.2 mpm) with a 40 F (4.4 C) cold wall, and 50 fpm (15.2 mpm) with a 27 F (−2.8 C) cold wall.

4. The floor surface, which did not lose or gain heat from below, was warmed to average temperatures exceeding the room-air temperature, so that the floor contributed convective heat to the room air as well as reradiated heat to cooler wall surfaces.

5. The heat emission from the ceiling panel and the heat absorption by the cold wall vary almost linearly with the temperature difference between these surfaces and the room air. The combined coefficient of heat transfer from heated ceiling panels was found to be about 1.2 Btu per (hr) (sq ft) (F deg temperature difference between ceiling panel surface and room air). Metric: 5.9 kilocal per (hr) (sq m) (C deg difference). The combined coefficient of heat transfer to the cold wall was found to be about 2.0 Btu per (hr) (sq ft) (deg F temperature difference between room air and cold-wall surface). Metric: 9.8 kilocal per (hr) (sq m) (C deg difference).

6. Globe-thermometer temperatures, at 30- and 60-in. levels (76.2 − 152.4 cm), 1 ft away from the cold wall, varied from about 1½ F deg (0.8 C deg) below room-air temperatures to 5½ F deg (3.1 C deg) above room-air temperature.

7. Mean radiant temperature at the 30- and 60-in. levels (76.2 − 152.4 cm) are in general higher than the room-air temperature, the elevation of the mean radiant temperature being greater for the colder wall conditions, because of the higher panel temperatures.

8. The directional mean radiant temperature measured at the 60-in. level when facing the ceiling was a value between the room-air temperature and the ceiling panel temperature, varying with the distance from the cold wall. Likewise, at the same level, the directional mean radiant temperature measured when facing the cold wall was a value between the air and wall temperatures, decreasing with the distance from the cold wall. Directional radiation measurements may be of considerable importance in future comfort studies.

9. Within the range tested and for the same total heat output, reducing the width and increasing the temperature of the heated ceiling panel adjacent to the cold wall did not appreciably alter the distribution of room-air temperatures in the occupied zone of the room. The narrower, warmer panels increased the globe thermometer temperatures, the mean radiant temperatures, and the DMRT near the cold wall.

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APPENDIX

CALCULATION OF SURFACE TEMPERATURES

The calculations of glass surface temperatures for comparison with the observed temperatures were made as follows: The thermal conductance between the room-side surface of the glass and the surface of the cold wall behind the glass was determined as the quotient of the heat flow into the surface of the glass, Fig. 14, and the temperature difference between these 2 surfaces. This was done systematically for heights of 1-ft intervals. The conductance was found to be least at the top of the wall and averaged about

$$0.95 = \frac{\text{Btu}}{(\text{hr})\ (\text{sq ft})\ (\text{F})} \quad \text{or } 4.65\ \frac{\text{kilocal}}{(\text{hr})\ (\text{sq m})\ (\text{C})}.$$

Secondly, fictitious thermal conductances were determined for each 1-ft increment of cold-wall height, based on the heat transfer at that height as shown in Fig. 14, and the difference between the average-ceiling and the isothermal cold-wall temperatures. These conductances varied with the temperature difference and the position with respect to height above the floor. With the conductances known from ceiling panel to glass surface (from isothermal wall tests), and from glass surface to the cold surface behind the glass, the glass temperatures were calculated for the conditions of Test No. 513.

The calculations of surface temperatures shown by the other curves of Fig. 15 were made as follows: For any construction, the *U*-value shown in the Heating Ventilating Air Conditioning GUIDE gives the over-all heat-transfer coefficients. From this *U*-value the inside film conductance was deducted, and the conductance, as determined above, substituted. For fixed indoor (70F, 21.1C) and outdoor temperatures, the surface temperatures and proper inside coefficients were calculated by trial-and-error methods.

Lateral conduction of heat from a warm section of the wall to a cooler section was neglected in this procedure.

DISCUSSION

W. P. CHAPMAN, Milwaukee, Wisc. (WRITTEN): As is usually the case our Laboratory Staff at Cleveland has made a significant contribution based on a meticulous gathering and analysis of data. Although the authors have pointed out that the conditions, *should not necessarily be regarded as good practice for producing comfort*, and that, *a project will be undertaken . . . in the future to determine the effects of radiation . . . on human comfort*, I would like to comment further on the implication of several of the figures.

As the title suggests we are interested in cold wall effects in a ceiling-panel-heated room. Let us assume for the moment that we are considering these effects on an office worker located 3 ft from an outside wall; furthermore let us restrict our conditions to the 41.5 F cold wall and the 8-ft panel.

First, look at the thermal conditions around the worker's feet. From Fig. 3 it can be seen that the air temperature varies from 4.7 to 2.5 deg F below room air temperature, or for these tests from 65.3 to 67.5 F as the height varies from 2 to 12 in. above the floor. The velocity of this air reaches a maximum of about 35 fpm at a height of 3 in. (see Fig. 6). The directional mean radiant temperature as given in Fig. 10 for these conditions is about 53 F.

The condition is nearly the same for the area around the worker's shoulders (assume a height of 4 ft). Here the air temperature is 3 or 4 deg warmer—about 70 F—and the velocity is down below 25 fpm, so would be considered as still air. The directional mean radiant temperature, however, is essentially the same 53 F.

In other words, *localized* cooling is quite likely. It is true that these tests were not controlled for comfort, but it is also true that these *localized* conditions would not be corrected if air temperatures were brought up to 75 or even 80 F.

Now change these conditions and consider the office worker at a distance of 6 ft from the cold wall, these localized conditions have just about disappeared. Notice too that cold wall temperatures of 60 F do not present localized difficulties even as close as 2 feet. Normally, however, the engineer cannot do much about cold wall temperatures. Changing the outside temperature or the architectural features are equally difficult. There is, however, an opportunity to change the *average surface temperature*; simply add a small source of heat below the window. It is sufficiently accurate to use the average unheated surface temperature instead of the mean radiant temperature in determining the size and temperature of this wall panel.

As the authors have pointed out in conclusion No. 8, *Directional radiation measurements may be of considerable importance in future comfort studies.* I certainly believe that this statement is borne out by the evidence presented in this paper.

R. J. MINDAK, Chicago, Illinois, (WRITTEN): Mr. Schutrum is to be congratulated for producing another excellent, well organized, comprehensive study in the panel-heating, panel-cooling area of research. It should prove to be a very valuable companion to the previous papers. However, there are a number of points which the discusser would like to raise.

First, it should be emphasized that a decrease in cold wall surface temperature results in an increase in the vertical temperature gradient throughout the room. For example, for a cold wall temperature of 60.1 F, the average temperature gradient between the 6-in. and 90-in. level is approximately 4.2 deg at the center of the room; for the condition of a 41.5 F cold wall temperature, the vertical temperature variation between the 6-in. and 90-in. heights averages 14 deg; and for the 27.3 F cold wall condition, the gradient is 18.5 deg.

The authors' second conclusion is misleading. It is stated that for the worst conditions, the vertical gradient between the 2-in. and 60-in. levels is only 5.6 deg and that

under usual conditions it would be expected to be much less. The reviewer would like to point out that between the 6-in. level and the 72-in. level, which covers roughly a range from the ankles to the top of the head of modern man exists an average temperature gradient of 10.6 deg. This condition is undesirable for comfort in the discusser's opinion. How much this condition would be improved under actual conditions is debatable.

The data of Figs. 8 and 9 emphasize interesting results. It will be noted from Fig. 9 that for 12-ft wide panels the maximum variation in MRT with distance from the cold wall amounts to 9 deg, whereas for a 4-ft panel the maximum variation is 5 deg. This is for the condition of the 41.7 F cold wall temperature. Note that the highest mean radiant temperature for the 4-ft panel occurs in close proximity to the cold wall. For the 12-ft wide panel the mean radiant temperatures are lowest in the vicinity of the cold wall. However, Fig. 8 indicates that the maximum variation in Globe Thermometer readings is not significantly different for the different panel widths.

Fig. 11 indicates that in the half of the room away from the cold wall the DMRTs for the ceiling and the cold wall are in general closer together for the 4-ft panel than they are for either the 8-ft or 12-ft panel. It should also be pointed out that the DMRT for the cold wall is considerably higher for the 4-ft wide panel condition throughout most of the room. These data therefore indicate that the 4-ft panel may in general provide more uniform comfort throughout the room for very cold wall conditions. The disadvantage of the 4-ft panel is, however, the fact that at a distance up to 4 ft from the cold wall the top of one's head may feel excessively warm as indicated by the high DMRTs.

It was mentioned that agreement between calculations and experimental results for the heat rates were found to be in reasonable agreement. I am wondering if the authors would care to comment on the order of magnitude of the agreement, i.e., whether the agreement is ± 2, 5 or 10 percent. Also, are any data available showing variation in DMRT facing the ceiling *vs* distance from the ceiling?

In discussing Fig. 13 the authors indicate that the radiation exchange is probably largely responsible for the variation of cold wall heat flow gradients with the heated panel width and temperature. I would like to point out that this effect may be calculated using configuration factors and emissivities. I am wondering if such calculations were made and, if not, they should be to provide an interesting comparison between theory and experiment.

In conclusion the reviewer would like to support the authors' results by stating that in an almost identical study in an almost identical room, very similar results were obtained as reported briefly by the reviewer several years ago.

PRESTON MCNALL, JR., Minneapolis, Minn. (WRITTEN): The authors are to be commended upon this contribution to our information on radiant effects with cold walls. The paper also points up the need for evaluation of radiation as a thermal comfort variable, in regard to both MRT and DMRT. The data presented showed remarkably high MRT elevations above air temperatures. While, as pointed out in the paper, furnishings in the room would tend to reduce these elevations, the infiltration of outdoor air instead of 70 F air would require a panel temperature even higher, which would probably result in greater unevenness of MRT and DMRT with different positions in the room. Perhaps these high elevations are also caused by the limited amount of cold wall and the very well insulated neutral surfaces of the room.

I have taken the case shown in test No. 512 of the paper with a cold wall temperature of roughly 60 F. The corresponding panel temperature was 83.4 F with 70 F infiltration air. Assuming ½ air change per hour and a U value of 0.17 for the wall, the outdoor temperature would have to be about -30 F. This would result in an additional 2520 Btu per hr load on the panel, or an additional panel temperature increase from 83.4 to about 90 F for the 12-ft panel. For this calculation the data of the paper were used for panel outputs.

I have also plotted MRT *vs* air temperature at the 30-in. and 60-in. levels for tests 510 and 511 of the paper. These are shown in Fig. A along with some field data taken by myself in a typical modern residence at about 0 F outside temperature. The MRT values were obtained with a Two-Sphere Radiometer of our own design. This instru-

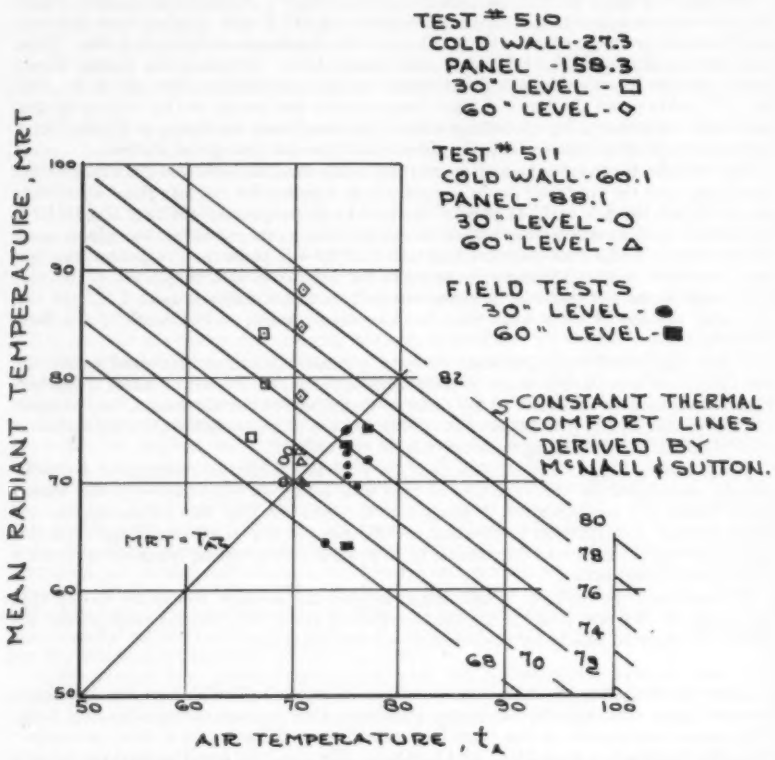

FIG. A—CROSS PLOT OF DATA FROM AUTHORS' PAPER WITH SOME COMPARATIVE FIELD DATA

ment was described in *A Two-Sphere Radiometer*, by Sutton and McNall, ASHAE TRANSACTIONS 1954, p. 297. Also shown is a thermal comfort correlation made by the discusser and David Sutton by analysis of the available literature data.

Air velocity and humidity effects are assumed average and constant for this purpose. These thermal comfort lines assist in analyzing the expected non-uniformity of thermal comfort existing in the various cases. DMRT effects are not included. Notice that for the cases of the paper and the field data the non-uniformity in air temperature is not excessive at the 30-in. and 60-in. levels. However, the MRT variation is in all cases

greater than the air temperature variations, particularly in the field data and the severe test No. 510. The field data shows an 11 deg spread in MRT and test 510 shows a 14 deg spread. Translating this into thermal comfort spreads according to Fig. A, it shows the residence experienced an 8 deg spread in comfort level while the test 510 showed 10 deg spread and test 511, a 2.5 deg spread.

Our field experience has indicated that in the heating season MRT readings, except at isolated unusual locations are never as far above air temperature as the paper reports but do vary from below to above over a range greater than the air temperature for panel systems. The residence plotted is not a panel system and in these cases MRT values are almost always below air temperature as indicated in Fig. A. Thus, this particular problem is not at all unique with panel systems and our experience indicates it is more severe with other systems, although our surveys have been limited and the structures and weather conditions were not comparable.

F. J. LINSENMEYER, Detroit, Mich., (WRITTEN): This paper is a valuable contribution to the knowledge of ceiling radiant panel heating. By heating the ceiling at decreasing depths from the outside wall, more uniform temperatures may be obtained.

TABLE A—DMRT FACING CEILING

Test No. 507 (Center of Room)

HEIGHT ABOVE FLOOR, IN.	DISTANCE FROM COLD WALL-INCHES				
	12	36	76	108	130
60	98.1	104.4	108.0	96.4	87.5
30	91.9	93.1	97.7	93.4	88.0
12	88.7	89.1	92.8	89.8	85.7

Air velocities and temperature down the wall, as reported, qualify under the definition as a *non-draft* condition. Floor surface temperature at a distance one foot from the wall exceeds the air temperature just above it by 6 deg. I have found this relation is about 3 deg when a concrete slab on ground without edge insulation, but at 3 ft from outside wall.

The 5-ft level ambient temperatures are about 70 F and measured by thermocouples *not affected by radiation*. This is probably the optimum for comfort and compares with the Globe Thermometer at 75 F. It is my belief that a thermostat setting should be somewhere between the two.

Higher absorption rates are obtainable when solar loads are reflected from windows or venetian blinds to ceiling. By proper interpolation it is believed Fig. 12 curves will verify this—if we will turn this room 90 degrees so that the cold wall assumes the position of the ceiling.

Panel temperatures up to 171 F were used in these tests. I have observed a panel at that temperature and I believe it is considered comfortable by the owners.

AUTHORS' CLOSURE (Mr. Schutrum): Mr. Mindak has asked two questions. One was in regard to the order of the agreement between calculated and observed heat flows through the cold wall. The average difference between calculated and observed heat flow rates was 6 percent, but was only about 3 percent for the 5 tests having cold wall surface temperatures of 42 F and colder. The radiation exchange was calculated using the surface temperature of the cold wall, and the area-weighted-average temperature of the ceiling, floor, and other 3 walls. Equation 8, reference 14, for convection heat transfer to walls, was used to calculate the convection component.

Mr. Mindak asked if data are available showing the variation in DMRT facing the ceiling *vs* distance from the ceiling. Typical data are shown in Table A.

Mr. Mindak also discussed the effect of radiation configuration factors on the variation of heat flow through the cold wall. Configuration factors for the radiation interchange between the ceiling panel and the cold wall vary as much as 4 to 1, *i.e.* radiation heat flow from the ceiling to the cold wall may be 4 times as great near the top of the wall as in a corner location on the wall, near the floor.

With regard to Dr. McNall's comments, the purpose of the air introduction into the test room was to provide ventilation, and the ventilation air was brought in at room-air temperature, which was 70 F. The ceiling was therefore the only source of heat. If outdoor air had been used instead of the 70 F ventilation air, the panel area or temperature would have had to be increased.

The limited amount of cold wall and the well insulated surfaces of the room, as Dr. McNall discussed, may possibly cause the mean radiant temperatures to be higher than those he has measured in the field. On the other hand, from the data reported in the paper, the required ceiling temperature would be higher and therefore a greater variation in directional radiant temperatures would exist if the cold wall areas were greater and the other walls were un-insulated.

Dr. McNall's Fig. A showing the correlation of mean radiant temperature and air temperature was interesting and helpful, and we urge him to present this research as a paper. The slide shows an 8 deg change in mean radiant temperature is equivalent to a 10 deg change in the room air temperature. It shows the importance of the mean radiant temperature on human comfort.

Dr. McNall has plotted the data from the tests having the extreme cold wall temperature, 27 F. Had the data from the 40 F cold wall tests been plotted, and I believe this is a more representative temperature, the spread in the comfort levels would exist between the test room data and that observed in a residence.

Dr. McNall and his co-workers have developed a very fine instrument in the two-sphere radiometer which he used to obtain the data in the residence. Not long ago one of these instruments was presented to the Society's Research Laboratory, and in the future, measurements of mean radiant temperatures will be greatly facilitated.

I wish to thank Mr. Lensenmeyer and Mr. Chapman for their valuable contributions to the paper.

1599

No. 1599

PERFORMANCE OF COVERED HOT WATER FLOOR PANELS

Part II — Room Conditions

By E. L. SARTAIN*, RACINE, WIS., AND W. S. HARRIS**, URBANA, ILL.

IN AN EARLIER paper[1] the Floor Slab Laboratory was described. In Part I of this paper[2], details were set forth covering (*a*) the heating installation, (*b*) the nature of the floor coverings used and their method of installation, (*c*) the instrumentation of the laboratory, and (*d*) the operating conditions maintained during the tests. The data presented here were all obtained in this same laboratory, using the same equipment and the same methods of testing.

ROOM-AIR TEMPERATURE AND VELOCITY DISTRIBUTION

Air temperatures were measured in each test room at levels 3, 12, 30, 60, 84 and 93 in. above the floor. At the center of the room, the maximum temperature difference between the various levels was obtained by subtracting the lowest temperature from the highest irrespective of distance above the floor. These are correlated with indoor-outdoor temperature difference in Table 1. About 35 tests were made in each case with indoor-outdoor temperature differences ranging from 63 to 15 F. A statistical analysis of the data showed that at 95 percent confidence level the average difference in room air temperature from 3 in. above the floor to 3 in. below the ceiling in Rooms A and B and at indoor-outdoor temperature difference of 80 F would not exceed 2.3 F. For Rooms C and D this variation would not exceed 3.8 F. Interchanging the floor coverings in Rooms C and D did not affect the air temperature difference.

The correlation of room-air temperatures as measured at the 30-in. level in the center of the room with the corresponding indoor-outdoor temperature difference

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** Research Professor of Mechanical Engineering, University of Illinois. Member of ASHAE.
[1] Exponent numerals refer to References.
Presented at the 63rd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Chicago, February, 1957.

are shown for each room in Table 1. The trend was for the room-air temperature to decrease slightly as the indoor-outdoor temperature difference increased. However, observations of the room-air temperature near the thermostat indicated that indoor-outdoor temperature difference had no effect on the room-air temperature at this location. Apparently the lower air temperature at the center of the room resulted from the circulation of cool air from the window which was not warmed to the normal room temperature.

Two days were selected in which average outdoor temperatures were about 37 F and 20 F, respectively, and the test points were very close to the average values from Table 1. Floor to ceiling temperatures at the center of the room were plotted for those days for all 4 test rooms (Fig. 1).

TABLE 1—AVERAGE ROOM-AIR TEMPERATURES[a] AND DIFFERENCES[b]

INDOOR-OUTDOOR TEMP DIFF	ROOM A		ROOM B	
	$t_c - t_f$	t_a	$t_c - t_f$	t_a
20	0.9 ± 0.2	71.1 ± 0.2	1.0 ± 0.1	71.1 ± 0.6
40	1.3 ± 0.1	70.6 ± 0.1	1.3 ± 0.1	71.1 ± 0.3
60	1.7 ± 0.2	70.1 ± 0.2	1.6 ± 0.2	71.1 ± 0.7
80	2.0 ± 0.3	69.6 ± 0.3	1.9 ± 0.3	71.1 ± 1.2
	ROOM C		ROOM D	
	$t_c - t_f$	t_a	$t_c - t_f$	t_a
20	0.5 ± 0.1	71.3 ± 0.5	1.2 ± 0.2	71.6 ± 0.2
40	1.5 ± 0.1	70.4 ± 0.2	1.9 ± 0.1	71.4 ± 0.1
60	2.5 ± 0.2	69.6 ± 0.5	2.7 ± 0.2	71.2 ± 0.2
80	3.5 ± 0.3	68.7 ± 0.9	3.4 ± 0.4	71.0 ± 0.4

Note: Figures in table are for confidence limit of 95 percent.
(a) t_a = Air temperature at center of room 30 in. above floor.
(b) $t_c - t_f$ = Maximum air temperature difference in center of room, 3 in. below ceiling to 3 in. above floor.

All room-air temperatures were corrected to a condition of 70 F at the 30-in. level for comparative purposes. Variations in temperature from floor to ceiling were small, and, except for floor-surface temperature, were virtually independent of outdoor temperature. Since the thermocouple at the 30-in. level in Room A was within a few inches of the thermo-integrator (surface temperature about 80 F), the indicated temperature at this point was probably high. Had the thermo-integrator not been present, probably the temperature curve in Room A would have more closely approximated that of Room B.

In Rooms C and D the temperature curves show (Fig. 1) a somewhat different shape from those in Rooms A and B. In Room C there was an increase in air temperature from the 3-in. level to the 30-in. level, while in Room D there was a slight decrease in temperature from the 3-in. level to the 12-in. level. The temperature at the 30-in. level in Room D was 0.6 deg to 1.2 deg higher than that 12 in. above the floor. Above the 30-in. level in all test rooms the room-air temperature was uniform for both moderate and low outdoor temperatures. When the outdoor

temperature was 20 F, the room-air temperature 3 in. above the floor in Rooms A and B at a location 2 ft from the north wall was about 3 deg lower than the corresponding value at the center of the room; whereas in Rooms C and D the difference in air temperature at these locations was 1.5 deg to 2.5 deg lower. This was to be expected since a definite downward movement of cold air was noticeable at the north wall. This cool air continued to move across the floor toward the center of the room.

To determine qualitatively the velocity of air movement in the rooms, a heated wire anemometer capable of measuring air velocities as low as 10 fpm was used as a probe. Fig. 2 represents a cross-section of Room C with air speed, direction and temperature indicated. The air was cooled at the north wall and window, dropped to the floor, and then moved in the direction of the south wall with a comparatively

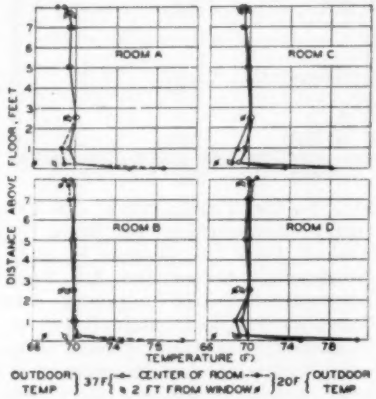

FIG. 1—ROOM AIR TEMPERATURE PROFILES

high velocity. Passing along the floor the air was heated by the panel whereupon it rose and drifted back slowly toward the north wall. Circulation of air could occur only as a result of air temperature differences. The air temperatures were uniform throughout the room except for locations along the floor and north wall. Near the window and within 12 in. of the floor, velocity was high and temperature was low. In extreme cases this combination of high velocity and low air temperature will result in undesirable drafts.

ROOM-AIR TEMPERATURE CONTROL

To evaluate the ability of the heating system to control the room-air temperature at a given value, the maximum and minimum air temperatures occurring at the location of the thermostat during 24-hr periods were recorded. Plotting indicated no correlation between observed maximum-minimum temperature difference and indoor-outdoor temperature difference; therefore, the data were grouped and averaged according to operating conditions (Table 2).

The ability of the thermostat to control the room-air temperature at a given value depends upon the thermostat differential, the response of the panel (time required for the panel to heat and cool), the time lag for transient heat flow through the exposed walls and windows, and changes in the outdoor temperature. The thermostat differential setting and the outdoor temperature for any one test period was the same for all 4 test rooms. The time lags for transient heat flow through the north exposure in Rooms C and D were probably less than for Rooms A and B because of the larger window area. Because of this and the fact that the carpeted

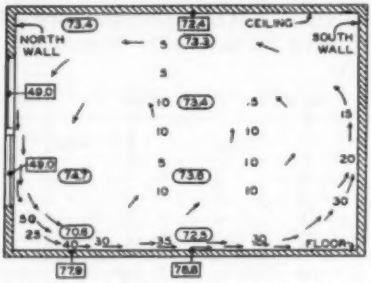

FIG. 2—AIR TEMPERATURE AND VELOCITY DISTRIBUTION IN ROOM C, WITH OUTDOOR TEMPERATURE OF 31 F. NUMBERS IN OVALS ARE AIR TEMPERATURES; THOSE IN RECTANGLES ARE SURFACE TEMPERATURES, BOTH IN FAHRENHEIT DEGREES. OTHER NUMBERS REPRESENT AIR MOVEMENTS IN FPM. ARROWS REPRESENT DIRECTION OF AIR MOVEMENT

floor panels did not respond to load changes as quickly as did the bare panels, room-air temperatures in Rooms C and D were subjected to more variation than in Rooms A and B.

Effects of floor coverings and operating conditions on room temperature variations are best shown by a room-by-room analysis of Table 2. Columns 2 and 4 show that the thermostat in Room A was apparently set about 1.5 deg higher during the tests in Case IV than in Case III. Col. 6 shows that the average temperature variation in this room was about the same in both cases. Columns 3, 5 and 7 show the standard deviations (SD) from the average values given in columns 2, 4 and 6, respectively. The standard deviation is a measure of the reproducibility of observations. Sixty-eight percent of all observations will normally be within ± one standard deviation of the average of all observations and 95 percent will be within ± two standard deviations. In Room A the standard deviations were about the same in Case IV as in Case III. Thus, it is apparent that while there was about a 1.5 deg change in the thermostat setting, the control of room-air temperature was not affected. In Room A the daily range of variation in room-air temperature at the thermostat was 0.5 to 1.3 deg for Case III and 0.6 to 1.6 for Case IV.

A similar analysis in Room B shows that the standard deviations in Cases III and IV were about the same while the thermostat setting was about 1.5 deg F higher for Case IV than for Case III. Thus, as in Room A, the control of room-air temperature was not affected by small changes in the setting of the thermostat resulting from the inability of the observer to reproduce identical settings from test to test. The average maximum minus minimum air temperature differences in

TABLE 2—ROOM-AIR TEMPERATURES

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)[a]
CASE	AVG MAX TEMP., F	SD (MAX)	AVG MIN TEMP., F	SD (MIN)	AVG MAX − MIN, F	SD MAX − MIN	VARIATION IN ROOM AIR TEMP., F
Room A							
III	72.2	0.5	71.3	0.3	0.9	0.4	0.5–1.3
IV	73.9	0.6	72.8	0.3	1.1	0.5	0.6–1.6
Room B							
III	72.7	0.6	70.5	0.6	2.2	0.6	1.6–2.8
IV	74.5	0.5	71.7	0.7	2.8	0.7	2.1–3.5
Room C							
I	74.6	0.6	73.0	0.3	1.6	0.6	1.0–2.2
II	76.0	0.8	74.0	0.4	2.0	0.8	1.2–2.8
III	73.5	1.3	70.9	0.5	2.6	1.4	1.2–4.0
IV	74.6	1.0	72.1	0.6	2.5	1.2	1.3–3.7
V	75.0	2.0	70.6	0.6	4.4	2.1	2.3–6.5
Room D							
III	74.3	1.2	71.0	0.5	3.3	1.3	2.0–4.6
IV	75.8	0.9	72.3	0.5	3.5	1.3	2.2–4.8
V	75.4	2.8	69.4	0.5	6.0	2.7	3.3–8.7

[a] Col. 6 + Col. 7

Operating Conditions:

Case I—Jan. 25–Feb. 30, 1952, Room C— Bare Panel.

Case II—Mar. 1–30, 1952, Room C—Bare Panel.

Case III—Dec. 2–31, 1952, Room A—Rubber tile; Room B—Asphalt Tile; Room C—Heavy Carpet and Pad; Room D— Light Carpet and Rubber Pad.

Case IV—Jan. 3–26, 1953, Room A—Rubber Tile; Room B—Asphalt Tile; Room C—Heavy Carpet and Pad; Room D—Light Carpet and Rubber Pad.

Case V—Feb. 27–Mar. 29, 1953, Room C—Light Carpet and 32 oz. Pad; Room D— Heavy Carpet and Rubber Pad.

Room B were higher than in Room A indicating a difference in the thermostat operating differentials even though the differential settings of the thermostats were the same. This affected the daily variation of room-air temperature in Room B which was 1.6 to 2.8 deg F for Case III and 2.1 to 3.5 deg F for Case IV (Col. 8).

The table shows that in Room C, Cases I and II, the standard deviation for the maximum room-air temperature was 0.6 to 0.8 deg F, while that for the minimum temperature was 0.3 to 0.4 deg F. During this time no floor coverings were used in the room. The daily range of room-air temperature at the thermostat was 1.0 to 2.2 deg F for Case I and 1.2 to 2.8 deg F for Case II. The effect of the carpet and pad in Room C, as shown by Cases III and IV, was to increase the standard deviation to values of 1.0 to 1.3 deg F for the maximum and 0.5 to 0.6 for the mini-

mum. As shown in column 8, this resulted in a large daily variation in room-air temperature.

The values for the standard deviations in Room D were very nearly the same as those in Room C. However, the average maximum minus minimum room-air temperature was about 0.7 to 1.0 deg F higher, indicating a larger variation in daily room-air temperature. The standard deviations from the average maximum room-air temperature in Rooms A and B for these periods of time were only about one-half as large as those in Rooms C and D. The standard deviation from the average minimum room-air temperatures in Rooms A and B were nearly the same as those in Rooms C and D.

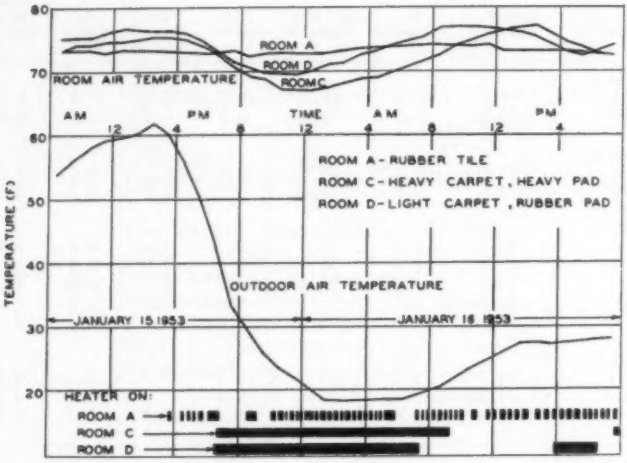

FIG. 3—CONTROL OF ROOM AIR TEMPERATURE WITH RAPID CHANGES IN OUTDOOR TEMPERATURE

Any change causing an increase in the values of Col. 5 of Table 2 will increase the variation in minimum air temperature occurring in the room at the end of the off-periods of the thermostat, thus making it necessary to increase the thermostat setting accordingly to insure that the room-air temperature remains above a predetermined minimum value. Changes causing an increase in the values of Col. 3, Table 2 will increase the normal variations in maximum room-air temperature occurring at the end of thermostat on-periods. Such overruns in room-air temperature are usually due to heat storage within the panel. However, the rate of heat loss from the room resulting from differences in construction or differences in outdoor temperature may be a contributing factor. For example, when the carpets were interchanged in Rooms C and D the value in Col. 7 for Room C was increased even though the carpet was lighter. This was due to the warmer weather encountered during Case V than during Case IV. As the values in Col. 7 increase, room-air temperature becomes more erratic. Thus, for Case II, Room C, the

normal maximum minus minimum room-air temperature difference was between 1.2 and 2.8 deg (Col. 8); whereas, in Case V it was between 2.3 and 6.5 deg.

One can conclude from Table 2 that the addition of carpets and pads over bare floor panels increased both the average maximum minus minimum room-air temperature difference and the standard deviation from this average. The maximum room temperatures (Col. 2) and standard deviations (Col. 3) were affected the most. In other words, other factors remaining constant, the addition of floor coverings to bare floor panels will reduce the ability of the system to maintain a

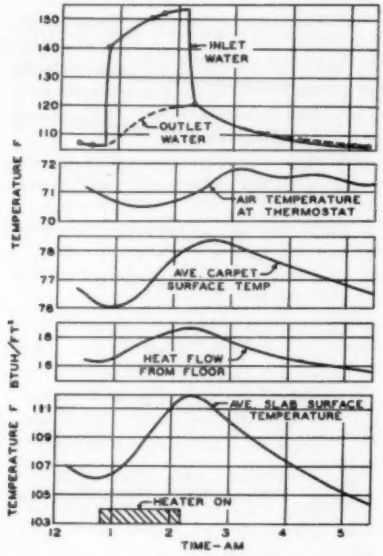

FIG. 4—SPECIAL STUDY OVER ONE HEATER CYCLE IN ROOM C, MADE FOR JANUARY 10, 1953 WITH AVERAGE OUTDOOR TEMPERATURE OF 33 F

constant room-air temperature. The greater the thermal resistance of the floor covering, the greater the resulting room-air temperature variation.

In order to show the effect of floor coverings on room-air temperature control with large outdoor temperature changes, a day was chosen during which there was a sharp decrease in outdoor temperature. Fig. 3 shows a plot of room-air and outdoor temperature *vs.* time for Rooms A, C and D for this day.

The room-air temperature for Room A (rubber tile) remained relatively constant even during the rapid change in outdoor temperature. However, such was not the case in Rooms C and D (carpet and pad). The thermostats started the heating systems at about 6:45 p.m. in Rooms C and D. Even so, the room-air tempera-

tures continued to decrease until about 11:00 p.m., January 15. At this time they leveled off at a value some 3 to 4 deg lower than the thermostat setting and then began to climb. Heat was finally being supplied at a great enough rate to more than offset the increased room heat loss due to the outdoor temperature drop. The minimum room-air temperature occurred in Room C about one hour later than in Room D.

By 7:30 a.m. the thermostat in Room D was satisfied, but the thermostat in Room C continued to call for heat until about 9:15 a.m. At 8:30 a.m. and 2:30 p.m. the air temperatures in Rooms C and D reached their respective maximum

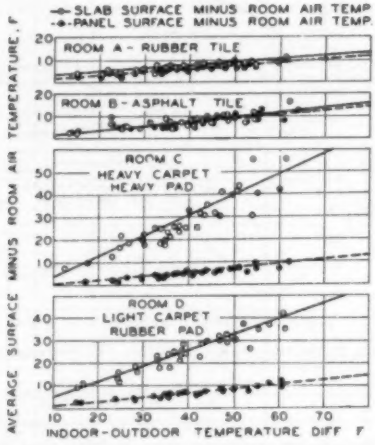

FIG. 5—PANEL AND SLAB SURFACE MINUS ROOM-AIR TEMPERATURES

values. While the maximum room-air temperature in Room D occurred one hour after the heater went off, the maximum room-air temperature in Room C occurred about 5 hr after the heater went off due to the increased heat storage in the panel and the less severe exposure in Room C. After the outdoor temperature became relatively constant, the room-air temperature in all test rooms became more stable. Much better control of room-air temperature was afforded in Room A than in Rooms C and D due to the comparatively rapid response of the system.

These curves show the effects of the carpet and pad combinations on the control of the room-air temperature. Part of the difference in the control characteristics of Rooms A, C and D might be attributed to the differences in exposure. However, additional panel capacity was provided in design to offset this portion of the difference. Observations in Rooms A, C and D as reported in Table 2 verify the conclusion that the additional exposure had a negligible effect on the control as compared to the effect of the carpet and pad combinations.

Since, during the study previously discussed, there was an extremely long heater operating period for Room C due to the rapid decrease in outdoor temperature,

the overrun in room-air temperature was exaggerated. Therefore, a typical day was chosen to make a study over one complete heater cycle in Room C. The outdoor temperature during this study was relatively constant at about 33 F. Temperatures of slab surface, carpet surface, room-air and water were measured during this test along with heat flow from the slab to the carpet. The floor covering consisted of the heavy weight carpet and 40 oz pad. Fig. 4 shows a plot of the measured temperatures and heat flows *vs.* time. At 12:30 a.m. the room-air temperature at the thermostat was dropping. At approximately 12:50 a.m. the thermostat called for heat, and the water heater was turned on. At this time the room-air temperature was 70.8 F. The panel inlet water temperature almost immediately increased from 106 F to 140 F, and the average slab surface temperature began to increase above 106 F. At the same time the average heat flow from the slab to the floor covering began to increase. The average carpet surface temperature continued to drop until about 10 min after the heater turned on, whereupon it started to increase. Meanwhile, the room-air temperature at the thermostat continued to decrease and reached its minimum value at about 1:25 a.m., or about 35 min after the heater went on.

The panel inlet water temperature continued to rise until about 2:10 a.m. when the thermostat was satisfied and the heater went off. The room-air temperature at this point was again about 70.8 F. Five or 10 min after the heater turned off, the average slab surface temperature reached a maximum, and thereafter decreased along with the heat flow from the slab to the floor covering. Even though the heat flow from the slab to the floor covering was decreasing the average carpet surface temperature continued to rise until about 2:40 a.m. when it reached a maximum. The maximum carpet surface temperature occurred 30 min after the thermostat was satisfied. The room-air temperature continued to rise until about 3:10 a.m. when it reached a maximum. There was about an hour's delay between the time when the water temperature reached its maximum value and the room-air temperature reached its maximum. There was a 25-min time delay between the time when the slab surface and the carpet surface reached their maximum values.

This test indicates that there was a relatively long period between the times when the thermostat called for heat and when the air temperature started to increase. The effect of the carpet and pad was to retard the flow of heat from the water to the room air and cause about 0.3 deg under and a 1.0 deg overrun in room-air temperature. After the thermostat called for heat it was satisfied 1 hr 20 min. Thus, the response time for the heating system was very long.

ROOM SURFACE TEMPERATURES

Floor Covering and Slab Surface Temperatures: Floor surface temperatures were obtained by thermocouples installed on the surface of the floor coverings, while the slab surface temperatures were obtained from thermocouples located on the slab surface. Fig. 5 shows a plot of floor covering surface (panel surface) minus room-air temperature difference *vs.* indoor-outdoor temperature difference. Also plotted in the same figure are the slab surface minus room-air temperature differences. The dotted curve of Fig. 5 shows that at design conditions the panel surface temperatures in all test rooms were from 12 to 15 F above the room-air temperature measured in the center of the room at the 30-in. level.

In Rooms A and B the temperature drop across the tile was very small as evidenced by the differences in the values of the 2 curves for each room. Such was not the case in Rooms C and D. At design conditions the temperature drop across the carpet and pad in Room C was about 54 F, while the temperature across the carpet and pad in Room D was about 38 F. These relative values would be expected since the effective conductance of the carpet and pad in Room C was greater than that in Room D.

The equivalent thermal resistance of the floor covering in each room is given by Equation 2 in Part I[2] of this paper. The value of Q/A at design conditions for each room was obtained by multiplying the percent above floor heat flow from Fig. 7, Part I, by the total heat flow from the panel in Fig. 6, Part I, at 80 F indoor-outdoor temperature difference, and dividing by the panel area.

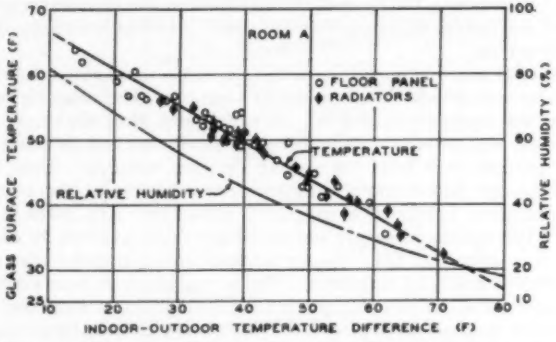

FIG. 6—INSIDE WINDOW SURFACE TEMPERATURE AND ALLOWABLE RELATIVE HUMIDITY INDOORS

Since for Room C the upward heat flow chosen from Part I of this paper included that for the inactive panel area while the temperature difference across the floor covering was evaluated for only the active panel area, the heat flow (taken from Fig. 6, Part I) had to be corrected to include the active panel only. Recorded data indicate that the total upward heat flow should be reduced by about 5 percent to correct for the inactive panel area. Therefore, the equivalent thermal resistance of the floor covering in Room C was

$$54.0/(0.95)(36.8) = 1.55$$

while that in Room D was

$$38.0/40.5 = 0.94 \text{ hr (sq ft)(F) per Btu}$$

The thermal resistances of these carpet and pad combinations in Rooms C and D were 1.85 and 0.99 hr (sq ft) (F) per Btu, respectively, as determined in the standard guarded hot plate by the method described in Part I. The 2 thermal resistances determined by these different methods were within 5 percent in Room D, while those in Room C differed by about 16 percent. The differences in thermal

TABLE 3—ROOM SURFACE TEMPERATURES, FAHRENHEIT DEGREES

OUTDOOR TEMP	ROOM A		ROOM C	
	35.0	12.5	35.0	12.5
Surface Temp				
East Wall[a]				
24 in.	71.4	72.0	73.5	75.1
72 in.	70.3	70.3	70.5	70.0
West Wall[a]				
24 in.	70.5	70.1	70.9	71.2
72 in.	70.2	69.4	70.1	69.9
South Wall[a]				
24 in.	70.3	69.4	70.6	70.5
72 in.	70.4	69.7	70.3	69.7
North Wall[a]				
24 in.	67.7	65.4	66.8	63.3
72 in.	67.3	64.8	67.8	63.0
89 in.	—	—	68.6	67.8
N. Window[a]				
42 in.	52.0	39.7	51.2	37.4
51 in.	52.3	39.6	50.9	46.1
Floor[b]				
6 in.	77.4	82.1	73.1	75.9
12 in.	74.5	75.8	73.3	78.4
18 in.	76.5	80.1	74.5	77.7
24 in.	75.0	76.1	74.9	78.4
30 in.	77.0	79.6	75.4	78.6
36 in.	76.0	77.8	75.3	78.3
42 in.	77.8	80.9	75.4	79.0
54 in.	76.5	78.5	76.0	79.4
78 in.	77.5	80.2	75.6	78.7
96 in.	73.2	74.0	75.7	79.6
114 in.	69.8	69.0	76.0	79.4
132 in.	69.3	68.4	72.2	73.0
Ceiling[b]				
3 in.	67.6	64.5	—	—
12 in.	68.3	66.5	—	—
21 in.	68.3	67.1	—	—
30 in.	68.5	67.8	69.6	69.2
39 in.	68.0	68.4	—	—
48 in.	69.7	69.1	—	—
54 in.	69.7	69.5	—	—
63 in.	70.2	69.6	—	—
72 in.	70.2	69.5	—	—
87 in.	70.1	69.5	—	—
102 in.	69.8	69.1	70.3	70.7
120 in.	69.6	68.7	—	—

[a] Inches measured above floor.
[b] On N-S center line of room; figures are inches from North Wall, each case.

resistances of the floor coverings as determined by the 2 methods could be due to uncontrollable variables. The method of surface temperature determination in the 2 tests was different. Under actual operating conditions the heat flow from the slab to the room was not steady, and there was probably considerable heat storage within the carpet and pad. The moisture content of the floor coverings could have influenced the results to the extent that the thermal resistance would change and heat would be required to evaporate any moisture present in the covering. No attempt was made to evaluate the moisture content of the floor covering for any test. Furthermore, the samples used for hot plate tests were not necessarily representative of the actual coverings used since variations in materials and manufacturing techniques could result in differences in physical properties.

No comparison of the apparent thermal resistance and the hot plate thermal resistance for the tile in Rooms A and B was made because the temperature drop across the tile was very small. Since the curves were extrapolated to design conditions, the results would not have been very accurate.

Glass Surface Temperatures: Thermocouples installed on the inside surfaces of the windows in the test rooms provided a means of determining the glass surface temperatures. Fig. 6 shows a plot of the inside window surface temperature for Room A *vs.* indoor-outdoor temperature difference. This curve was identical with those obtained for all other test rooms. Also indicated are the test points obtained in the $I = B = R$ Research Home[3] with a conventional hot water heating system using small tube radiators. One curve could be drawn through all the points indicating that there was no difference in the observed window surface temperatures in the 2 cases. At design conditions of 80 F indoor-outdoor temperature difference the glass surface temperature was about 26 F.

The dotted curve in Fig. 6 represents the relative humidity of 72 F air which will result in a dew-point temperature equal to the measured surface temperature of single glazed windows.

Wall, Floor and Ceiling Surface Temperatures: Surface temperatures of the walls, floors and ceilings were obtained by thermocouples. For the unexposed walls in all rooms and the ceilings in Rooms C and D the average reading of all the thermocouples on one surface represented the area weighted average temperature of that surface.

To show the distribution of surface temperature in the test rooms, 2 days were selected; one when the outdoor temperature was about 12 F, and the other for an outdoor temperature of about 35 F. Table 3 shows the surface temperatures for various elements of Rooms A and C during these two days. The unheated surface temperatures were considerably less for the day when the outdoor temperature was 12 F than for the day when the outdoor temperature was 35 F.

Table 3 shows that at 24 in. above the floor in both Rooms A and C the east wall surface temperatures were greater than the west. This was due to the presence of the supply piping in the east walls of these rooms. The table also shows that the temperature of the lower half of the east wall was greater in Room C than in Room A because of the increased water temperature in the supply piping in Room C. In both rooms the temperature of the lower half of all the partitions was greater than that of the upper half, probably the result of conduction and radiation from the heated slab to the partitions.

Table 4 shows the relationship between room-air temperature minus surface temperature and indoor-outdoor temperature difference for all 4 test rooms. Average surface temperatures were subtracted from the room-air temperature for all

readings to eliminate the effects of fluctuations in room-air temperature. The room-air temperature was measured at the 30-in. level in the center of the room.

The difference between the air temperature and the AUST*, the average surface temperature (AST) and the average north wall surface temperature (NWST) are all shown in Table 4. The NWST does not include the window surface tempera-

TABLE 4—ROOM AIR TEMPERATURE T_a MINUS NWST, AUST, AND AST[a]

INDOOR-OUTDOOR TEMP DIFF F	ROOM A WITH RUBBER TILE		
	T_a MINUS NWST	T_a MINUS AUST	T_a MINUS AST
20	2.0 ± 0.3	1.0 ± 0.2	0.5 ± 0.2
40	4.0 ± 0.2	1.9 ± 0.1	0.8 ± 0.1
60	5.9 ± 0.4	2.8 ± 0.3	1.0 ± 0.3
80	7.9 ± 0.7	3.7 ± 0.5	1.3 ± 0.5
INDOOR-OUTDOOR TEMP DIFF F	ROOM B WITH ASPHALT TILE		
	T_a MINUS NWST	T_a MINUS AUST	T_a MINUS AST
20	2.1 ± 0.4	1.0 ± 0.3	0.3 ± 0.3
40	4.1 ± 0.2	1.9 ± 0.2	0.7 ± 0.1
60	6.0 ± 0.4	2.8 ± 0.3	1.0 ± 0.3
80	8.1 ± 0.7	3.8 ± 0.6	1.3 ± 0.5
INDOOR-OUTDOOR TEMP DIFF F	ROOM C WITH HEAVY CARPET AND HEAVY PAD		
	T_a MINUS NWST	T_a MINUS AUST	T_a MINUS AST
20	2.2 ± 0.4	1.0 ± 0.1	0.5 ± 0.3
40	4.1 ± 0.2	1.6 ± 0.1	0.2 ± 0.2
60	5.9 ± 0.5	2.3 ± 0.1	0.0 ± 0.4
80	7.8 ± 0.8	3.0 ± 0.3	−0.3 ± 0.6
INDOOR-OUTDOOR TEMP DIFF F	ROOM D WITH LIGHT CARPET AND LIGHT PAD		
	T_a MINUS NWST	T_a MINUS AUST	T_a MINUS AST
20	1.9 ± 0.2	0.9 ± 0.4	0.1 ± 0.3
40	3.8 ± 0.2	2.0 ± 0.2	0.1 ± 0.1
60	5.6 ± 0.3	3.0 ± 0.4	0.2 ± 0.3
80	7.4 ± 0.6	4.1 ± 0.7	0.2 ± 0.5

[a] Air temperatures measured at 30-in. level at center of room. Figures in table are for confidence limit of 95 percent (Average Values).

ture. In all rooms the north wall surface temperature at design conditions was about 8 deg F lower than the room-air temperature and in all test rooms except Room C, the AUST was about 4 deg F lower than the room-air temperature. In Room C the AUST was about 3 deg F lower than the room-air temperature. Since

* The AUST represents the area weighted average temperature of all the unheated surfaces of the room.

there was an additional window in Room C as compared to Rooms A and B, one would expect a lower AUST for the same room-air temperature. However, the effect of the carpet and pad was to warm the lower portions of the walls adjacent to the panel. Also, the supply piping was located within the partitions separating Room A from Room B and Room C from Room D. The higher water temperature required in Rooms C and D resulted in higher surface temperatures for the partitions between these 2 rooms. These caused an increase in the AUST which more than compensated for the additional window. The net effect was to cause a higher AUST than in Rooms A and B. In Room D the AUST was the same as in Rooms A and B, even though the exposure was much more severe. The effect of the carpeting and piping on the wall surface temperature in Room D affected the AUST by about the same amount as the additional window and exposed wall.

The AUST has been established as a parameter which determines the total heat transfer from a panel for a given panel surface temperature and infiltration rate.[4] Relationships between AUST, panel surface temperature, infiltration,

TABLE 5—MRT, AST AND AUST IN ROOM A FLOOR PANEL HEATING[a]

INDOOR-OUTDOOR TEMP DIFF F	MRT AT CENTER OF ROOM	AST	AUST
20	70.5 ± 0.1	70.1 ± 0.4	70.2 ± 0.2
40	70.0 ± 0.1	69.9 ± 0.1	68.9 ± 0.1
60	69.4 ± 0.1	69.6 ± 0.4	67.5 ± 0.2
80	68.8 ± 0.3	69.3 ± 0.7	66.2 ± 0.4

[a] Figures in table are for confidence limit of 95 percent (average values).

room-air temperature, and total panel heat output have been found at the ASHAE Laboratory. These relationships apply to rooms whose geometry is similar to that which would be encountered in practice.

Since calculation of the AUST does not include the heated panel area the net radiant heat exchange between a body located in the room and its surroundings would depend not only upon the AUST, but also upon the panel surface temperature. Furthermore, the location of the body with respect to the surrounding surfaces must be taken into consideration. *Body temperature depends upon the balance between heat production and heat loss.*[5] Heat production is determined by the metabolic rate, and heat is lost from the body by convection, evaporation, and radiation. Therefore, for a given degree of activity or metabolic rate, the body surface temperature is determined by the environmental factors affecting the heat exchange between the body and its surroundings. These factors are dry-bulb temperature, humidity, air motion, and surface temperature of the surroundings[6, 7].

If a person were sitting at rest in the center of the test room his body temperature would then be determined by these environmental factors. Upon moving away from the center of the test room toward the window, the net radiant heat exchange between the body and surrounding surfaces would change as a result of a change in configuration factor of the body with respect to the surrounding surfaces. The configuration factor of the body with respect to the exposed wall would increase while the configuration factor of the body with respect to the other room elements would decrease[8]. If the air temperature, humidity, and motion were the same near the window as at the center of the room, a lower body surface temperature would result in order that the total heat loss from the body would remain equal

to the metabolic rate. The magnitude of this change in body surface temperature would depend upon the change in net radiant heat transfer from the body. This is also affected largely by the type of clothing worn.

However, the important fact here is that a change in the net radiant heat transfer from the body would result, and, therefore, one constant such as the AUST or AST could not represent the MRT (mean radiant temperature) which is an index of the amount of radiation taking place between the body and its environment, referred to a uniform environment. The MRT is a function of location of the body within the test room.

In Rooms A and B the AST was about 1 F below the room-air temperature while in Rooms C and D the AST was essentially the same as the room-air temperature. These data show that even though Rooms C and D had more severe exposures, the AST was slightly higher for a given room-air temperature than in Rooms A and B. This was due to increased panel area and higher temperature of the lower partition surfaces because of the presence of the supply piping and carpeting.

Several instruments have been employed for the measurement of MRT in an environment[7], including the eupathescope, the globe thermometer, and the thermo-integrator. (See Bibliography-Measuring Instruments, at end this chapter). During these tests MRT was measured by a thermo-integrator installed in Room A at the center of the room with the center of the instrument approximately 30-in. above the floor. Table 5 shows the relationship of the AST to the indoor-outdoor temperature difference for Room A. The AST and MRT decreased slightly with increasing indoor-outdoor temperature difference while the AUST dropped off rapidly. It is evident from Table 5 that the AST closely approximates the MRT obtained at the center of Room A, 30-in. above the floor. It should be pointed out that such may not be the case for other locations.

Since the AST approximated the MRT at the center of the room, the data of Table 4 suggest that a lower room-air temperature might be tolerated in Rooms C and D than in Rooms A and B because of higher AST's for a given outdoor temperature.

HEAT FLOW AND FLOOR SURFACE TEMPERATURE PROFILES

The heat flow and floor surface temperature profiles for bare floor panels have been reported in a previous paper[1]. Installation of tile over the panels in Rooms A and B had no significant effect on these characteristics.

Fig. 7 shows a plot of floor surface temperature and heat flow profiles in Rooms C and D with carpets and pads installed over the panels. For these tests the average carpet surface temperature increased toward the center of the room, but the heat flow from the panel to the carpeting was greatest near the exposed wall and decreased toward the center of the room.

In general, the effects of the carpeting were to cause an increase in the floor surface temperature along a line toward the center of the room and to smooth out the heat-flow profile from the panel to the carpet. The carpeting also caused some increase in the subfloor heat flow over that obtained for bare panels. The effect of the carpeting on the below floor heat losses was discussed in Part I of this paper.

CALCULATED AND MEASURED PANEL HEAT OUTPUT

The total heat output from the panel to the test rooms is equal to the sum of the radiation and convection components. The heat transferred from the panel by

convection and radiation was calculated using the equations presented in THE GUIDE 1955, pp. 569–571. The convection portion of the total heat transfer from the panel is given by[9]

$$q_c = 0.81\ (t_s - t_a)^{1.12} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

and the radiation is given by[10]

$$q_r = 0.142\ [(T_s/100)^4 - (T_w/100)^4] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

and

$$q_t = q_c + q_r$$

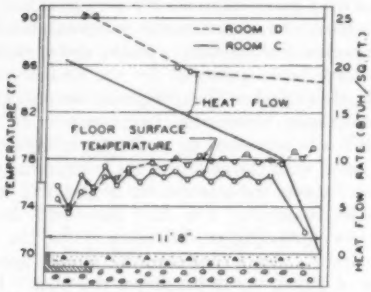

FIG. 7—HEAT FLOW AND SURFACE TEMPERATURE PROFILES IN ROOMS C AND D. DATA FOR DECEMBER 3, 1953 WITH OUTDOOR TEMPERATURE OF 29 F; ROOM AIR TEMPERATURE OF 70.5 F (BOTH ROOMS); AUST OF ROOM C AT 68.5 AND AUST OF ROOM D AT 68.8 F

where

q_c = heat transfer by convection in Btu per (hour) (square foot).
q_r = heat transfer by radiation in Btu per (hour) (square foot).
q_t = heat transfer, total, in Btu per (hour) (square foot).
t_s = panel surface temperature in Fahrenheit degrees.
t_a = air temperature in Fahrenheit degrees.
T_s = panel surface temperature in Fahrenheit (absolute).
T_w = average unheated surface temperature of the surrounding surfaces in Fahrenheit (absolute).

The air temperature used in these calculations was measured at a distance of 3-in. above the floor at 2 locations—2 ft from the north wall and at the center of the room. The air temperatures at these locations depended on outdoor temperature and differed from one another 2 to 4 deg F. The AUST was used for the temperature of the surroundings as recommended in THE GUIDE. Table 6 shows the relationship between the measured and calculated panel outputs for all test rooms. In all cases, the measured panel output was consistently higher than the calculated

panel output. At design conditions of 80 F indoor-outdoor temperature difference, the percent difference between the calculated and measured panel output was from 7 to 18 percent (based on the measured output). The percent difference was greater for the rooms with more severe exposures.

The calculated convection heat transfer from the panel was based on the equation expressing the heat flow from a horizontal flat plate facing upward with no unusual end effects[9]. The presence of the exposed wall in the test rooms caused a downward cold air current which passed over the surface of the panel as indicated in Fig. 2. This increased air velocity over the surface of the panel probably resulted in a greater heat flow from the panel by convection for a given surface to air temperature difference than that calculated by Equation 1. This could account for part of the difference in the calculated and measured panel output. Also, the

TABLE 6—CALCULATED VS. MEASURED HEAT FLOW TO ROOMS[a]

CALCULATED HEAT FLOW TO ROOM, BTUH	AVERAGE MEASURED HEAT FLOW TO ROOM, BTUH	
	ROOM A	ROOM B
1000	1010 ± 61	1225 ± 84
2000	2085 ± 62	2220 ± 84
3000	3160 ± 141	3220 ± 166
4000	4230 ± 230	4215 ± 262
5000	5305 ± 320	5210 ± 360
	ROOM C	ROOM D
1000	1225 ± 125	1310 ± 222
2000	2385 ± 81	2370 ± 135
3000	3547 ± 149	3430 ± 116
4000	4705 ± 250	4490 ± 185
5000	5870 ± 357	5550 ± 286

[a] Figures in table are for confidence limit of 95 percent.

panel surface temperatures measured in Rooms C and D may not have been accurate because of the difficulty in establishing the actual surface of the carpeting. The heat flow as determined by heat flow meters in Rooms C and D was actually the heat flow from the slab to the carpet and pad. Since there was some heat storage in the floor coverings, the heat flow from carpet to room at any instant was not necessarily the same as that measured by the heat flow meters. This may have been a factor influencing the difference in the measured and calculated panel outputs.

The relationship between panel output and panel minus room-air temperature is shown in Table 7. In Rooms A and B for which the exposures were similar, the panel outputs for a given panel minus room-air temperature difference were approximately the same. The outputs for Rooms C and D where the exposures were more severe were 15 to 20 percent greater than in Rooms A and B at the same panel minus room-air temperature difference. The same panel minus room-air temperature difference in all rooms does not represent the same outdoor conditions since the variation in panel surface minus room-air temperature with outdoor tempera-

ture was not the same in all test rooms (see Fig. 5). Variations in panel output were due to variations in exposure, infiltration, and panel area.

At design conditions the panel output for Room A was about 30.4 Btuh per sq ft while the output of the panel in Room B was about 35.4 Btuh per sq ft—a difference of about 5.0 Btuh per sq ft. These panel outputs may be obtained from Table 7 using panel surface minus room-air temperature corresponding to design conditions as obtained from Fig. 5. Based on the panel area, the difference in panel outputs represents about (116 x 5.0) or 580 Btuh. Calculations indicate

TABLE 7—AVERAGE PANEL HEAT OUTPUT TO ROOMS[a]

PANEL SURFACE TEMP. MINUS ROOM AIR TEMP., F	PANEL OUTPUT TO ROOM, BTUH PER SQ FT	
	ROOM A	ROOM B
2	4.2 ± 1.2	5.2 ± 0.9
4	9.5 ± 0.7	10.2 ± 0.6
6	14.7 ± 0.5	15.3 ± 0.5
8	19.9 ± 0.9	20.3 ± 0.6
10	25.1 ± 1.5	25.3 ± 0.9
12	30.4 ± 1.7	29.5 ± 1.3
14	35.6 ± 2.8	35.4 ± 1.6
	ROOM C	ROOM D
2	7.1 ± 0.8	6.1 ± 1.3
4	12.5 ± 0.6	11.3 ± 0.9
6	17.8 ± 0.5	16.4 ± 0.6
8	23.2 ± 0.8	21.6 ± 0.7
10	28.6 ± 1.2	26.7 ± 1.0
12	34.0 ± 1.5	31.9 ± 1.4
14	39.3 ± 2.0	37.0 ± 1.9

[a] Figures in table are for confidence limit of 95 percent.

that a difference of 0.3 air changes per hr could account for this difference in panel output, but no measure of the infiltration rate was made during these studies.

Data have been presented[4] which correlate AUST, panel output, panel surface temperature and infiltration. However, no comparison of results could be made because infiltration rate was not measured or controlled in the studies reported here. Therefore, the room-air temperature was lower, and panel heat output greater for a given test as compared to those reported by ASHAE for no infiltration.

RELATIVE HUMIDITY

The relative humidity in each test room was obtained with calibrated hair-type relative humidity indicators. Table 8 shows the relationship between relative humidity and indoor-outdoor temperature difference. A comparison of the relative humidity in Rooms B and D for Test I indicates that during mild weather (20 F indoor-outdoor temperature difference) the relative humidity was about the same for both rooms. However, as the indoor-outdoor temperature difference

increased, the relative humidity in Room B decreased more rapidly than in Room D. The relative humidity in Room D for Test II was higher than for either room during Test I. Also there appeared to be more scattering of test points.

Throughout the entire test period it was observed that especially during moderately cold weather preceded by damp, warm weather, condensation occurred on the windows in rooms with carpeting, while the windows in rooms which had no carpeting were dry. This may have been due to the tendency of the carpet to absorb moisture when it was raining and to release it later as the slab was heated when the outdoor temperature dropped. However, the fact that the relative humidity in rooms with carpeted panels was consistently higher than that in rooms

TABLE 8—AVERAGE RELATIVE HUMIDITY IN TEST ROOMS[a]

INDOOR-OUTDOOR TEMP. DIFF. F	MEASURED RELATIVE HUMIDITY IN PERCENT		
	ROOM B WITH ASPHALT TILE	ROOM D WITH LIGHT CARPET AND RUBBER PAD TEST I	ROOM D WITH HEAVY CARPET AND RUBBER PAD TEST II
20	32.3 ± 6.5	34.4 ± 2.2	33.9 ± 2.3
40	23.7 ± 2.3	30.6 ± 0.8	30.8 ± 1.2
60	15.1 ± 6.8	26.9 ± 2.0	29.8 ± 3.2
80	6.5 ± 12.7	23.1 ± 3.8	29.5 ± 5.1

[a] Figures in table are for confidence limit of 95 percent.

with uncovered panels even during extended cold weather suggests that there may have been an actual increase in the rate of transfer of water vapor through the concrete floor slabs in the carpeted rooms.

Tests[11] conducted with laboratory specimens of various construction at the Forest Products Laboratory have indicated that the rate of migration of water vapor through floor slabs is proportional to the vapor pressure difference across the slab. Furthermore, those tests indicated that the type of vapor barrier used under the concrete floors in the Floor Slab Laboratory was not very effective in reducing water vapor transfer rates.

In the test rooms with carpeting on the floor, it was necessary to operate the heating system with higher slab temperatures than in rooms without carpets. This in turn increased the temperature of the gravel fill under the slab. Considering the air saturated in the voids between pieces of stone constituting the gravel fill under the slab, calculations were made for the vapor pressure difference across the slab. Fig. 8 shows a plot of this vapor pressure difference across the slab for Rooms B and D. The vapor pressure difference in Room D was consistently greater than that in Room B. The smaller motive head for the transfer of vapor through the slab in Room B plus the presence of the asphalt tile which acted as an additional vapor barrier must have accounted for the difference in relative humidities between test rooms.

By establishing a mass balance on the water vapor in Rooms B and D, the resulting relative humidity in the test rooms was calculated for various outdoor temperatures. Condensation and absorption of water vapor by materials in the test rooms as well as the transfer of water vapor through partitions and ceilings were omitted from consideration. The resistance to transmission of water vapor by carpets and pads was considered negligible, and the room-air temperature was taken to be constant at 72 F. Under steady-state conditions the rate of water vapor transferred through the slab plus the rate of water vapor brought into the room due to infiltration was equal to the rate of water vapor removal from the room due to ex-

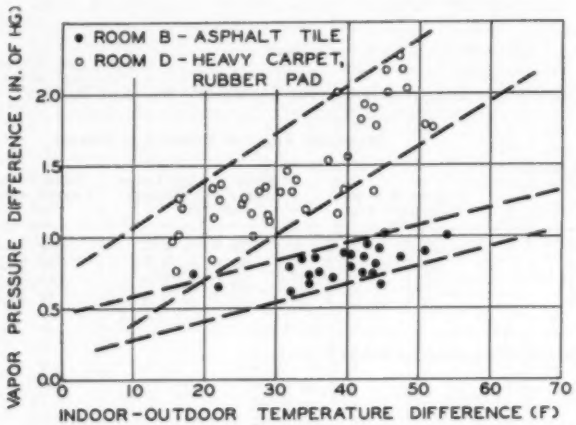

FIG. 8—VAPOR PRESSURE DIFFERENCE ACROSS SLAB (CALCULATED)

filtration plus the rate of water vapor transferred through the exterior wall. In equation form this mass balance becomes:

$$Ww_1 + M_s(P_s - P_w) \pm M_w(P_w - P_{wo}) = Ww \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where

W = infiltration rate, pounds per hr

w_1 = vapor content of infiltrating air, pounds of water vapor per pound of dry air

w = vapor content of exfiltrating air, pounds of water vapor per pound of dry air

P_s = saturation pressure of water vapor in the voids between pieces of stone constituting the gravel fill, inches mercury

P_w = partial pressure of the water vapor in the test room, inches mercury

M_s = equivalent permeance of the concrete floor slab, grains per hr (sq ft) (inches mercury)

M_w = equivalent permeance of the exterior wall, grains per hr (sq ft) (inches mercury)

For similar slab construction the value of M_s has been given[11] as $\frac{2.45 \text{ gal}}{\text{day (1000 sq ft) (0.345) psi}}$. Converting this value to appropriate units $M_s = 8.2$ grains per hr (sq ft) (in. Hg). The value of M_w was calculated from values reported in THE GUIDE and $M_w = 0.264$ grains per hr (sq ft) (in. Hg). Since the infiltration rate was unknown a value of ½ air change per hr was assumed for both test rooms. This was equivalent to 676 cu ft per hr or 50.7 lb per hr standard air. The mass balance for Room D can then be written:

$$50.7w_i + (8.2)(169)\,(P_s - P_w) \pm (0.264)\,(81)\,(P_w - P_{wo}) = 4354W\frac{P_w}{29.9 - P_w} \quad (4)$$

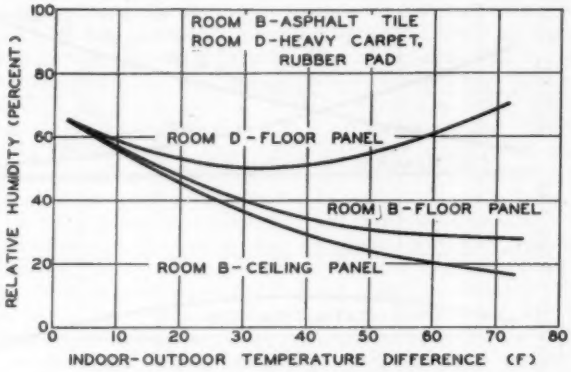

FIG. 9—CALCULATED RELATIVE HUMIDITY BASED ON MASS BALANCE OF WATER VAPOR

where

$$w = 4354\frac{P_w}{29.9 - P_w}$$

In order to determine the condition of the outdoor air, the relative humidity at a given dry bulb temperature was taken equal to that for a five year average obtained in Urbana, Illinois [12]. For a given outdoor temperature, the value of w_i was determined, and the value of P_s was taken as the saturation pressure corresponding to the observed temperatures under the slab. Thus, the equation was solved for P_w, and the relative humidity was determined for 72 F room-air dry-bulb temperature. Fig. 9 shows a plot of the indoor relative humidity thus determined *vs.* indoor-outdoor temperature difference. For Room B, the relative humidity with floor panel heating was consistently higher than that obtained with ceiling panel heating. This was due to the increased vapor transfer rate through the floor slab when it was heated. Recorded data in Room B show that the measured relative humidity with ceiling panels was actually less than that obtained with a heated floor panel which is in agreement with the calculated values of relative

humidity. For Room D, the calculated relative humidity was consistently much higher than that for Room B due to increased vapor transfer rates through the floor slab. These trends are in agreement with those indicated by Table 8. In Room D the relative humidity decreased with decreasing outdoor temperature to a minimum value and thereafter increased. Beyond the minimum value, the vapor transfer rate through the slab was great enough to more than offset the decreased vapor gain due to infiltration to such an extent that the relative humidity in the test room increased. Below the minimum value, the increased gain in vapor due

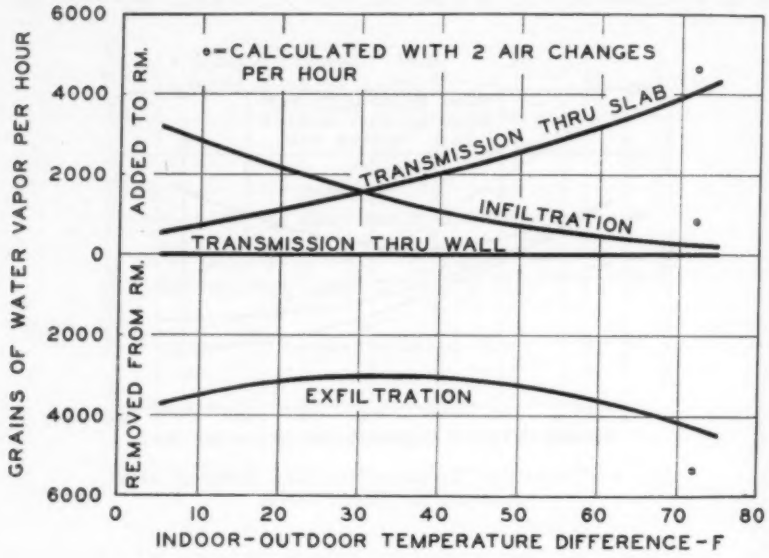

FIG. 10—CALCULATED WATER VAPOR MASS BALANCE

to infiltration of outdoor air with high moisture content more than offset the decreased vapor transfer rate through the slab so that the net result was an increase in the relative humidity in the test room.

A comparison of Table 8 and Fig. 9 shows that the measured relative humidity was never as high as the calculated. This could have been due to the omission of condensation and absorption terms in the vapor mass balance. Condensation on cold windows definitely occurred during cold weather as pointed out previously. Also, the assumption of 100 percent relative humidity in the voids of the gravel fill could have been in error. Previous work[13] has shown that the relative humidity depends upon the moisture content of the soil under the slab. All of these unaccounted for vapor losses would tend to reduce the relative humidity in the test rooms and bring the calculated and measured values into closer agreement.

The individual terms of Equation 3 representing vapor gains and losses have been plotted in Fig. 10. Vapor loss due to exfiltration (which was the only vapor

loss included in Equation 3) was equal to the sum of vapor gains due to infiltration and transmission through the slab and exposed wall. The transmission through the exposed wall is seen to be practically insignificant. Vapor gains due to infiltration decreased with decreasing outdoor temperature as a result of decreased moisture content of the outdoor air. Transmission through the slab increased with decreasing outdoor temperature as a result of increased vapor pressure differences across the slab. The sum of these 3 curves represents the vapor loss due to exfiltration.

In order to determine the infiltration rate, W, in equation 3, at zero outdoor temperature, the value of relative humidity as obtained from Table 8, Test I, Room D, was used. Solving Equation 4 for the infiltration rate resulted in a value of about 200 lb per hr or 2 air changes per hr. This value would seem high for infiltration at zero outdoor temperature. However, the W terms in Equation 3 include all the losses in vapor from the test room. Therefore, in the actual case, these terms include such factors as condensation and absorption. Using 2 air changes per hr, the values of each component of Equation 3 were plotted in Fig. 10. Both vapor gains due to infiltration and transmission through the slab were higher than those obtained with ½ air change per hr. Also, the vapor loss due to exfiltration was greater than the corresponding value with ½ air change per hr. The difference in vapor loss for the 2 cases was about 1050 grains per hr or 0.15 lb per hr. This amount of water vapor could possibly have been due to condensation, absorption, and unaccounted for losses in Equation 3.

During the 1953–54 heating season, the limit switch settings and the firing rates of heaters serving carpeted rooms were increased. This was done because the results of special pick-up studies indicated that the carpeted panels responded very slowly with limit switches set at 130 F and firing rates equal to 1.33 times the design heat loss. Also, the attic temperature was uncontrolled for the 1953–54 studies. Under these conditions during extremely cold weather the vapor pressure difference across the slab with a heavy carpet and heavy pad reached as high as 5.1 in. Hg. Thus, a large motive force was provided for vapor transmission through the slab. This resulted in even higher relative humidities in rooms with carpeted panels than in rooms with bare or tiled panels.

CONCLUSIONS

1. At design conditions of 80 F indoor-outdoor temperature difference the maximum difference in room-air temperature between the levels 3-in. above the floor and 3-in. below the ceiling was 3.5 deg F. At a location 2 ft from the north wall, the room-air temperature was 3 deg F lower than at the center of the room. Cooled air dropped to the floor at the north wall and window and moved in the direction of the south wall with a somewhat high velocity.

2. Addition of floor coverings to bare floor panels reduced the ability of the system to maintain a constant room-air temperature. The greater the thermal resistance of the floor covering, the greater the resulting room-air temperature variation. Carpets and pads retarded the flow of heat from the water to the room-air, resulting in poor response.

3. Glass surface temperatures measured with floor panel heating were the same as those obtained in the Research Home with conventional radiation.

4. The exposed wall surface temperature was about 8 deg F lower and the AUST was about 4 deg F lower than the room-air temperature measured at the center of the room 30-in. above the floor.

5. The AST in Rooms A and B was about 1 deg F below the room-air temperature while in Rooms C and D the AST was essentially the same as the room-air temperature.

6. The MRT as obtained in the center of Room A with a thermo-integrator was the same as the AST for all outdoor conditions encountered.

7. The effects of the carpeting were to cause an increase in the floor surface temperature along a line toward the center of the room and to smooth out the heat flow profile from the panel to the carpet.

8. At design conditions of 80 F indoor-outdoor temperature difference the measured panel output was from 7 to 18 percent greater than the calculated panel output. For a given panel minus room-air temperature difference the panel heat output to Rooms C and D, which had more severe exposures, were 15 to 20 percent greater than outputs in Rooms A and B.

9. The relative humidity in Room D which had a carpet and pad was consistently greater than that in Room B, indicating that there was probably an increase in the rate of transfer of water vapor through the concrete floor slab in this room. Trend curves obtained from calculations based upon a mass balance for the water vapor were in agreement with those obtained from measurements of relative humidity.

ACKNOWLEDGMENTS

This report concerns one phase of the panel heating research project which is part of the comprehensive steam and hot water heating investigation being conducted by the *Institute of Boiler and Radiator Manufacturers* in cooperation with the University of Illinois. Eventually the results of this research project will be compiled in an Engineering Experiment Station Bulletin.

The authors wish to acknowledge especially the assistance of research assistants Albert J. Wiegand and Liang-Neng Tao in obtaining and compiling data for this report. Also, acknowledgment is made of companies which contributed equipment for this study.

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† This publication also contains information concerning the Thermo-Integrator and its uses.

DISCUSSION

R. B. RICE, Raleigh, N. C., (WRITTEN): Fortunately, the human body adapts itself readily, in its quest for comfort, to a wide range of conditions insofar as air temperature, air velocity, and humidity are concerned. It is the function of the engineer, however, to control and minimize the fluctuation of those variables and to limit conjecture concerning the results which can be anticipated from a particular design to an absolute minimum through valid interpretation of adequate research data compiled through extensive tests and experimentation involving the actual conditions under which a human being would be exposed with this particular set of design conditions.

The authors are to be commended for the excellent research report and the conclusive data contained therein which will be of inestimable value to the design engineer and architect alike in the design of living facilities and housing in America. Obviously the researchers have selected the worst possible conditions for establishing their test criteria:

the bare room without decorations. With the addition of furniture, pictures, drapes, and other objects the picture will change sharply. However, unless there is a preponderance of furniture or hangings, the blanketing effect will result in more rather than less comfort.

Again, the data on heat liberation from the panel with rubber tile and asphalt tile compare favorably with values anticipated by the engineer and from meager data which is already available. However, the information involving the heavy carpet and heavy pad in Room C are of tremendous importance to designers.

The report makes clear a point which is often neglected by both engineers and architects—the fact that the manner in which a room is to be decorated or treated in general may have considerable influence upon the performance of the heating system designed for that room. For example, if the heated slab is to be covered with a heavy carpet and

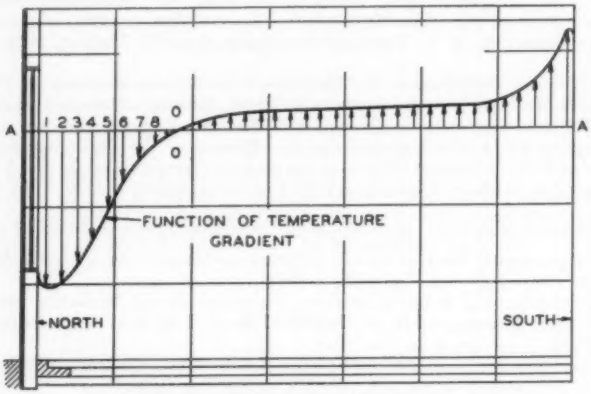

FIG. A—VELOCITY DISTRIBUTION AT A-A, ROOM C

heavy pad, it is evident then that the water in the heating coils must be capable of maintaining a temperature as much as 50 to 60 deg above that for a slab which is uncovered. Engineers have suspected this to be true but have never had adequate valid data until this time. Likewise, if a building is to be heated with floor panels and certain rooms are to be carpeted and others not, then an ample design allowance should be made at inception for this contingency.

Another important conclusion from the paper as revealed in Fig. 5 is the fact that the slope of the average slab surface temperature difference plotted against outdoor temperature is a straight line.

Referring to Fig. 3, it is suggested that if and when a hot water floor panel design of this type is installed with heavy carpeting and in a part of the country where the outdoor air temperature may recede below 30 F then some suitable augmenting device should be incorporated into the design to take care of the 16-hr lag or delay in the system recovering from rapid outdoor cooling. This can be partially compensated by designing a thermostatic control which would automatically adjust itself in anticipation of rapid decrease in temperature as well as wind velocity which also will sharply influence this type of heating system. A suitable augmenting device would be an electric heating unit or heating coils placed in the periphery of the room or in the walls.

It is pointed out that an increase in ceiling height will probably have a marked effect upon the curves in Fig. 1 and the general data in Fig. 2. Whether an unusually high ceiling would result in more pronounced velocities or more stratification of temperature, referring again to Fig. 2, is something that should be ascertained from analysis of additional and extensive test data.

The general conclusions from this excellent paper would certainly reveal the tremendous potentialities of the hot water floor panel for heating. It also reveals the fact that literally any type of floor covering can be employed without seriously affecting the heating of this space. Whether this would be true for a large room of several thousand square feet floor area would depend upon better understanding of some basic fundamentals.

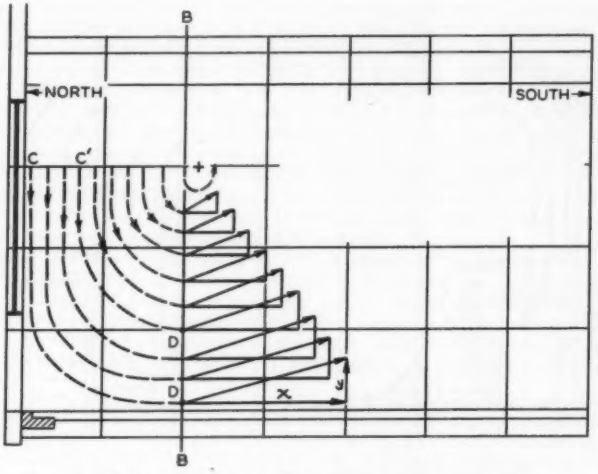

FIG. B—IDEAL FLOW EQUILIBRIUM CONDITIONS, ROOM C

For example, Fig. A of this discussion is a hypothetical velocity distribution curve drawn for the room in Fig. 2. The summation of the vectors to the left of the neutral point 0–0 would have to balance the summation of the vectors to the right. The areas under these curves represent the energy which is required for pumping the air, or circulating it, from the cold surface in the North wall across the heated surface in the floor and then vertically back to the starting point.

The vectors 1 through 8 are functions of the air velocity or motion. Were the outdoor and the inside temperature the same, and there were no heat disbursment in the room, then all vectors would reduce to zero and equilibrium would exist. Fortunately for this system, as the North wall becomes colder, the vectors nearer that wall become larger and larger by proportion and functional with the difference in the outdoor temperature and inside air and the pumping action results.

A better and more exact knowledge of this phenomenon is desirable in extrapolating the data of this research report and in employing it for larger rooms, higher ceilings, or for rooms with decorations. Similarly, far more intense and rigorous analysis of the phenomenon which takes place along the floor surface is needed. The velocity which

the air particles acquire in passing along the surface of the North wall then turning and passing across the heated surface of the floor will be influenced by a vertical component *y* resulting from the rate at which heat is added to the air particles by convection, conduction, and radiation (Fig. B). The *y* or vertical component would tend to compress the air above that region in the remainder of the room. This air in general will tend to resist compression, however, the net result will be a general upward motion. This action will not be complete, however, until the air has reached the opposite, or South wall.

In Fig. C an attempt is made to analyze the effect of placing furniture on the North and South walls of the rooms as this would affect the general flow lines of air motion. It is clear that the table, for example *g* on the North wall will in its immediate vicinity reduce the cooling effect on the air particles and therefore reduce the pumping action.

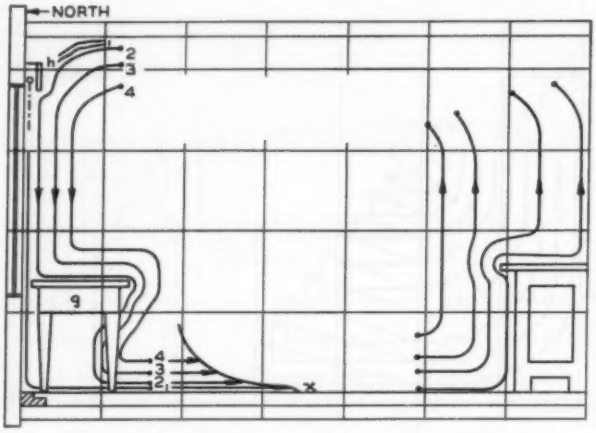

FIG. C—FLOW LINES WITH FURNITURE, ROOM C

However, that portion of the wall which is not covered by the table will have an increased burden.

For a small room, say 12-ft from wall to wall, a modest amount of furniture might have little or no effect upon ultimate comfort inasmuch as any reduction in air flow would compensate for decrease in heat transmission from the room. However, in large rooms a better understanding of actual flow patterns would be desirable in order to design adequately for ultimate comfort.

L. F. SCHUTRUM and T. C. MIN, Cleveland, Ohio (WRITTEN): During the last 5 years or so, research on panel heating has been carried on simultaneously in the Floor Test Laboratory at the University of Illinois and in the Society's Environment Laboratory in Cleveland. This offered an opportunity to compare data. Most comparisons have been good.

In this paper, the room air temperature profiles shown in Fig. 1 are similar to those measured in the Environment Laboratory. In those tests, the amount of variation in the air temperature in the first 2½ feet above the floor depended upon the degree of non-uniformity of the wall and ceiling surface temperature and the amount and temperature of the infiltration air.

The air motion in general was the same as shown in Fig. 2 of this paper. Air flows down the cold wall, across the floor and up the warmer opposite wall. The average surface temperature, AST, was about the same as the mean radiant temperature, MRT, in the center of the room at the 5-ft level.

Floor panel outputs for a given room air to panel surface temperature as reported in this paper are higher than those found in the ASHAE test room under uniform environment conditions. A 30-in. level air-temperature was used in the paper, whereas in the other, the 60-in. level was used. After adjusting for this difference, the outputs reported here for room A are about 10 to 20 percent higher. Non-uniform environments increase the output about 10 to 15 percent in the ASHAE tests, but on the other hand, room furnishings tend to reduce the difference of only 2½ percent from that of the uniform environment.

The Wilkes and Peterson's free convection equation, used in this paper, was formulated from data with large heated and cooled plates spaced only 4 inches apart. As a result, it gives higher values in heat flow than those reported by other investigators. Should other free convection equations be used, the difference between the calculated and measured values in heat flow would be even greater. However, air infiltration would cause additional convection. For 1 air change per hour at 0 F infiltration, the convection heat output for a heated floor panel at 85 F would increase about 40 percent.

The authors report a higher relative humidity in the room with the carpet which also has the higher concrete surface temperature and attribute this to a greater transfer of water vapor through the slab. It may be that when considerable vapor is transferred through the slab additional heat transfer to the ground below the slab is required to vaporize this water. An effective vapor barrier may then reduce the heat loss from the slab as well as improve the installation in other ways.

The authors also stated that for a given test the room-air temperatures were all lower in ASHAE tests as compared to those reported in this paper, and that as a result, the panel outputs reported here are greater for a given panel surface temperature and AUST than for those reported in ASHAE results. It seems to the discussers that if the air temperature at ASHAE tests is lower, the panel output would be greater for a given panel surface temperature and AUST. Enlightenment on this interpretation will be appreciated.

E. L. PERRINE, Evanston, Ill.: I would like to comment on the heat loss into the ground and relative humidity. Overall insulation below the floor slab will decrease the pickup time that was mentioned on the carpeted floor even if it does not give a significant fuel saving. In other words, resistance below will compensate for the resistance of the floor covering. Also, insulation below the slab will decrease the temperature in the soil below the vapor barrier and that will reduce the amount of moisture brought up through the concrete slab. Therefore, the insulation will give you 2 advantages in this case.

AUTHORS' CLOSURE (by Mr. Sartain): Mr. Rice's discussion of this paper certainly tends to broaden the scope of the results presented here. He is to be complimented for his unique qualitative analysis of the phenomena of air circulation. Situations which were posed involving air circulation in rooms having various geometries and furniture placement can be quantitatively answered only with the benefit of data from more comprehensive research.

Mr. Schutrum's and Mr. Min's discussion pointed out that floor panel heat outputs reported in this paper are higher than those found in ASHAE tests with uniform environments. It is felt that the reason for higher outputs obtained in tests at the University of Illinois may be due to the method used to measure average heat flow. Heat flow meters were installed on a room centerline perpendicular to a window as described in the paper *Heat Flow Characteristics of Hot Water Floor Panels*. Since equivalent local radiation and convection heat transfer coefficients are greater along the room centerline than at other areas, the average heat output of the heat flow meters times the total panel area would be greater than the true average total panel output.

A vapor barrier was installed under the slab as described in a previous paper. Ex-

periments performed by the Housing and Home Finance Agency have pointed out that the vapor barrier used under the slab in these studies was rather ineffective in retarding moisture transfer. This information is included in the paper as published in the TRANSACTIONS.

As the discussers point out, there is an apparent discrepancy between panel output, room-air temperature and average unheated surface temperature relationship as reported in the last paragraph under the heading *Calculated and Measured Heat Output.* This is the result of a printing error and is corrected as published in the TRANSACTIONS. Data have been presented at the ASHAE Laboratory which correlate the average unheated surface temperature, panel output, panel surface temperature, and infiltration. However, no comparison of results could be made because infiltration rate was not measured or controlled in the studies reported here. Therefore, the room-air temperature was lower and the panel heat output greater for a given test as compared to those reported by ASHAE for no infiltration.

Mr. Perrine's comments regarding heat loss into the ground and relative humidity agree with observations made at the Floor Slab Laboratory. However, no supporting data are available to substantiate these comments except that ground temperatures were only slightly lower immediately under the slab when sub-floor insulation was used. It is possible that vapor pressure under the slab would then be somewhat reduced.

1600

No. 1600

THERMAL DESIGN OF WARM WATER CONCRETE FLOOR PANELS

Prepared by a subcommittee* of TAC on Panel Heating and Cooling† and Members of the staff of The ASHAE Research Laboratory.

THIS PAPER presents a simplified procedure for the thermal design of water heated floor panels for use in residences and commercial buildings. It complements the previously published paper[1].

The procedures in both papers are based primarily on the experimental data obtained at the ASHAE Research Laboratory under the guidance of the ASHAE Technical Advisory Committee on Panel Heating and Cooling.† This work has been reported in a series of research papers which are listed in the references, together with other papers which contain supplementary data.

A panel designed by this procedure will maintain the desired room air temperature for the selected outdoor conditions. Room air temperature is the selected criterion of comfort, and the design procedure is restricted to situations in which the area-weighted average temperature of the walls, the ceiling, and glass does not differ greatly from room air temperature. The room-scale tests, which simulated various conditions of construction and outdoor temperature, showed that this near-equality of the 2 temperatures normally prevails.

Design Steps

Panel design requires specification of the following: panel area, size, spacing and depth of bury of pipe or tube, insulation on the underside of the panel, and the temperature drop and the mean temperature of the water. The procedure, summarized, is as follows:

1. Calculate the hourly rate of heat loss for each room.
2. Determine the available area for the panel in each room.

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* *Personnel*: R. L. Maher, chairman; W. P. Chapman, H. T. Gilkey, P. B. Gordon, E. F. Snyder, Jr., and J. M. van Nieukerken.

† *Personnel*: P. B. Gordon, chairman; W. S. Harris, vice chairman; A. B. Algren, W. P. Chapman, J. Everetts, Jr., H. T. Gilkey, E. S. Howarth, L. N. Hunter, A. T. Jorn, W. C. Kadow, G. D. Lain, H. A. Lockhart, C. O. Mackey, R. L. Maher, S. K. Smith, E. F. Snyder, Jr., W. F. Spiegel, J. M. van Nieukerken, G. L. Wiggs.

[1] Exponent numerals refer to References.

Presented at the 63rd Annual Meeting of the American Society of Heating and Air-Conditioning Engineers, Chicago, February 1957.

3. Calculate the required unit panel output.
4. Determine the required panel surface temperature.
5. Select type, size and spacing of pipe or tube and the insulation for the underside of the panel.
6. Determine the required mean water temperature.
7. Design the fluid circuit (panel piping and mains) and determine the size of the heating equipment, following conventional practice.

RANGE OF APPLICATION

Outside design conditions: Temperatures as low as −30 F.
Room air temperature: 70 to 76 F.
Air changes: not more than 2 air changes per hour.
Room dimensions: Rooms having normal proportions; ceiling height between 7 and 12 ft.
Room construction: Any type of wall construction and any amount of glass area. (Both, however, have an effect upon comfort.) Conventional interior finishes and furnishings.

PROCEDURE FOR HOT-WATER CONCRETE FLOOR PANELS (SLAB-ON-GRADE)

Step 1—*Heat Loss*: Calculate the heat loss of each room, following the recommendations contained in THE ASHAE GUIDE, but do not include any heat loss through the area covered by the panel.

Example: Three Rooms, A, B, and C, are to have a common water supply temperature; that is, they represent a single zone. They are to be maintained at 72 F air temperature when the outdoor air temperature is zero F. The floors in the 3 rooms are covered with heavy wall-to-wall carpeting, without a pad. Insulation which has a conductance of 0.4 Btu/(hr) (sq ft) (F) is placed along the edge of the slab and extends downward 2 ft below it. Room dimensions, exposure, and calculated heat losses are as follows:

ROOM	DIMENSIONS FT	EXPOSED WALL FT	HEAT LOSS BTU/HR
A	11 × 12 × 8	23	3300
B	11 × 12 × 8	12	2500
C	15 × 21 × 8	21	8000

Step 2—*Required Panel Output*: Divide the heat loss of each room by the maximum floor area in the room which can be used as a heating panel. The result is the minimum heat output per square foot of panel which will satisfy the requirements of the room. (The panel which requires the highest output per square foot will generally control the design, because the temperature of the water in the system must be high enough to produce the required output from that panel.)

ROOM	AVAILABLE PANEL AREA SQ FT	REQUIRED PANEL OUTPUT (q_u) BTU/(HR) (SQ FT)
A	132	25.
B	127	19.7
C	306	26.1

Step 3—*Panel Surface Temperature*: From Fig. 1, find the panel surface temperature needed to yield the required heat output to each room, using the output determined in Step 2 and the design room air temperature. This use of Fig. 1 is illustrated in Fig. 1A.

Floor panel surface temperatures in excess of about 85 F are not recommended because of the probability of discomfort to the feet. If the required heat output cannot be obtained from an 85 F floor panel, heat losses should be reduced or supplementary heating should be provided.

ROOM	PANEL SURFACE TEMPERATURE (t_p)
A	85. F
B	82.5 F
C	85.3 F

Step 4—*Downward and Edgewise Heat Flow*: 1. Determine the temperature of the surface of the concrete slab (t_s) by adding to the panel surface temperature, the temperature difference caused by the thermal resistance of the floor covering, if any:

$$t_s = t_p + (q_u \times r_{uc})$$

The value of (r_{uc}) for various floor coverings is given in Table 1.

ROOM	t_p	q_u	r_{uc}	t_s
A	85. F	25.	0.8	105 F
B	82.5 F	19.7	0.8	98.3 F
C	85.3 F	26.1	0.8	106.2 F

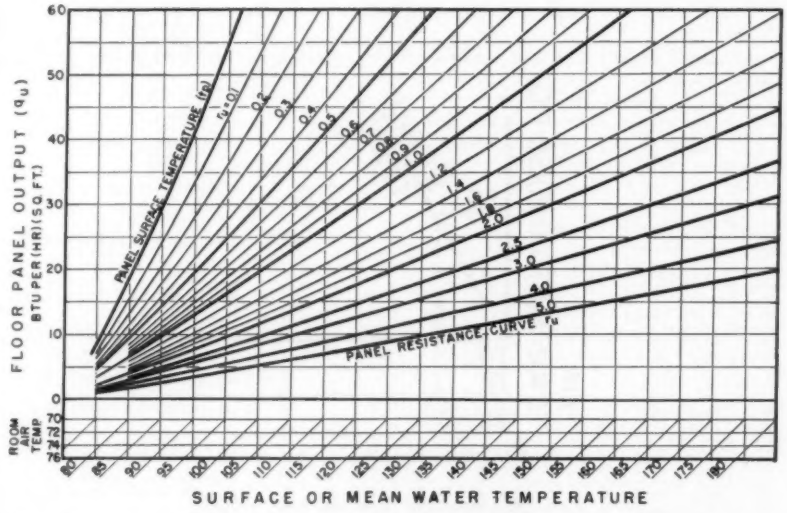

FIG. 1—FLOOR PANEL DESIGN GRAPH—PANEL SURFACE TEMPERATURE AND MEAN WATER TEMPERATURE VS. OUTPUT UPWARD

TABLE 1—THERMAL RESISTANCE OF FLOOR COVERINGS

DESCRIPTION	RESISTANCE, r_{us} (F) (HR) (SQ FT)/BTU
Bare concrete, no covering	0.00
Asphalt tile	0.05
Rubber tile	0.05
Light carpet	0.6
Light carpet with rubber pad	1.0
Light carpet with light pad	1.4
Light carpet with heavy pad	1.7
Heavy carpet	0.8
Heavy carpet with rubber pad	1.2
Heavy carpet with light pad	1.6
Heavy carpet with heavy pad	1.9

2. Determine the downward and edgewise heat loss coefficient (h) from Fig. 2 for the insulation to be used.

Insulation with a conductance of 0.4 Btu/(hr)(sq ft)(F) extended 2 ft below the slab results in a slab downward and edgewise heat loss coefficient of 0.97 Btu/(hr)(ft)(F).

3. Apportion the downward and edgewise heat loss uniformly across the panel as follows:

$$q_{de} = P \times h\ (t_s - t_{so})/A_p$$

ROOM	P	$(t_s - t_{so})$	A_p	q_{de}
A	23	105 F	132	17.7
B	12	98.3 F	127	9.0
C	21	106.2 F	306	7.1

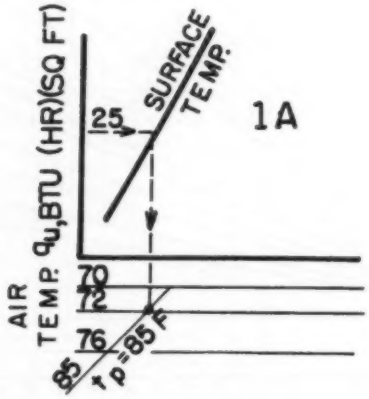

FIG. 1A—SHOWING USE OF FIG. 1 TO DETERMINE PANEL SURFACE TEMPERATURE (STEP 3)

Step 5—Upward Panel Resistance: Assume a tentative pipe or tube size and a spacing for each panel, choosing closer spacings and larger pipe or tube when higher heat outputs are required. From Table 2, find the thermal resistance (r_{us}) of the slab of each panel, using heat flow ratios (q_u/q_{do}) calculated from the flow rates determined in Steps 2 and 4, interpolating as required. To the slab resistance (r_{us}), add the resistance of the floor covering (r_{uc}, Step 4) to obtain the panel resistance to upward heat flow (r_u).

ROOM[a]	PIPE OR TUBE SPACING	q_u/q_{do}	r_{us}	r_{uc}	r_u
A	9	25/17.7 = 1.41	0.47	0.8	1.27
B	12	19.7/ 9.0 = 2.19	0.55	0.8	1.35
C	9	26.1/ 7.1 = 3.68	0.40	0.8	1.20

[a] All rooms: ½ in. pipe or ¾ in. tube; 4 in. concrete slab.

Step 6—Mean Water Temperature: For the required panel output (q_u) found in Step 2, the panel resistance (r_u) found in Step 5, and the room air temperature (t_a), find the mean water temperature (t_{mw}) from Fig. 1. This use of Fig. 1 is illustrated in Fig. 1B.

ROOM	t_{mw}
A	116.5
B	109.0
C	116.5

TABLE 2—THERMAL RESISTANCE OF BARE CONCRETE FLOOR PANELS

PANEL CONSTRUCTION	SPACING, IN.	THERMAL RESISTANCE							
		HEAT FLOW RATIO q_u/q_d OR q_u/q_{do}							
		1		3		5		10	
4 IN. CONCRETE SLAB—2 IN. COVER		UP r_{us}	DOWN r_{ds}	UP r_{us}	DOWN r_{ds}	UP r_{us}	DOWN r_{ds}	UP r_{us}	DOWN r_{ds}
½ in. (nom.) non-ferrous tube	9	0.57	0.52	0.46	0.84	0.43	1.17	0.42	1.97
	12	0.73	0.68	0.58	1.16	0.54	1.65	0.51	2.86
½ in. (nom.) ferrous pipe or ¾ in. (nom.) non-ferrous tube	9	0.49	0.42	0.41	0.66	0.39	0.90	0.38	1.80
	12	0.63	0.55	0.50	0.93	0.48	1.30	0.46	2.35
6 IN. CONCRETE SLAB—2 IN. COVER									
½ in. (nom.) non-ferrous tube	9	0.59	0.70	0.47	1.05	0.45	1.39	0.43	2.25
	12	0.78	0.90	0.60	1.40	0.56	1.97	0.54	3.21
¾ in. (nom.) non-ferrous tube	9	0.51	0.61	0.43	0.87	0.41	1.13	0.40	1.78
	12	0.68	0.78	0.54	1.23	0.51	1.63	0.49	2.61
¾ in. (nom.) ferrous pipe	9	0.47	0.55	0.40	0.77	0.39	0.98	0.38	1.50
	12	0.63	0.71	0.50	1.07	0.48	1.44	0.46	2.36
1 in. (nom.) non-ferrous tube or 1 in. (nom.) ferrous pipe	12	0.59	0.66	0.48	0.98	0.46	1.30	0.45	2.11
	15	0.73	0.83	0.57	1.21	0.54	1.73	0.51	2.74

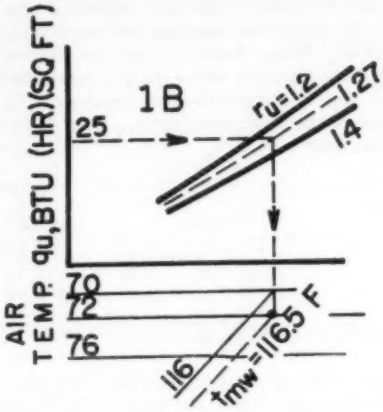

FIG. 1B—USE OF FIG. 1 TO DETERMINE MEAN WATER TEMPERATURE (STEP 6)

Step 7—Design Mean Water Temperature: Select a single design mean water temperature (t_{dmw}) for each group of rooms which is to comprise a zone, choosing the highest mean water temperature (t_{mw}) of the group.

In the *example*, 116.5 F is chosen.

Step 8—Design Panel Output: From Fig. 1, find the panel output (q_u) for design mean water temperature (t_{dmw}), room air temperature (t_a), and panel resistance (r_u). Fig. 1C shows how Fig. 1 is used in this step.

Also from Fig. 1, find the design panel surface temperature. If it exceeds 85 F (see Step 3) in a room, go back to Step 5 and choose a wider spacing or smaller pipe or tube for that room.

ROOM	(q_u)	t_p
A	25.	85. (unchanged)
B	24.0	84.5
C	26.1	85.3 (unchanged)

Step 9—Design Panel Area: Divide the room heat loss found in Step 1 by the design panel output found in Step 8.

ROOM	A_p
A	132 (unchanged)
B	104
C	306 (unchanged)

Step 10—*Total Panel Output*: Add the apportioned downward and edgewise heat flow (q_{de}) to the design panel output (q_u) and multiply by the design panel area to obtain the total panel output. If the design panel output is appreciably different from the panel output (q_u) used in Steps 3 and 4, the apportioned downward and edgewise heat flow (q_{de}) should be redetermined.

ROOM	q_u	q_{de}	$q_u + q_{de}$	A_p	$A_p(q_u + q_{de})$
A	25.0[a]	17.7[a]	42.7	132	5636
B	24.0	11.6[b]	35.6	104	3702
C	26.1[a]	7.1[a]	33.2	306	10,159

[a] Unchanged. [b] Redetermined.

Step 11—*Fluid Circuit*: Design the fluid circuit (panel piping and mains) for a temperature drop of 10 F to 20 F between the water inlet (t_i) and outlet (t_o) of the panel (see THE ASHAE GUIDE, Chapter on Hot Water Heating Systems).

Step 12—*Boiler Size*: Size the boiler according to method contained in THE GUIDE, Chapter on Heating Boilers, Furnaces, and Space Heaters. The net Btu rating of the boiler should equal or exceed the total output of all panels plus any other loads on the boiler.

PROCEDURE FOR HOT WATER CONCRETE FLOOR PANELS (INTERMEDIATE SLAB)

Step 1—*Heat Loss*: Follow the procedure for slab-on-grade construction, Step 1.

Step 2—*Required Panel Output*: Follow the procedure for slab-on-grade construction, Step 2.

Step 3—*Panel Surface Temperature*: Follow the procedure for slab-on-grade construction, Step 3.

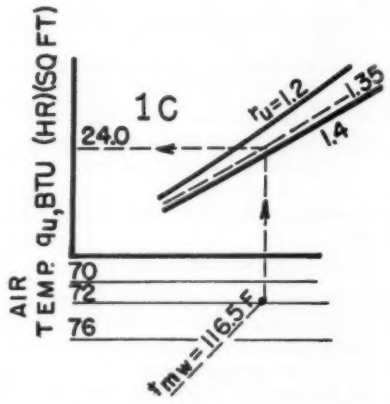

FIG. 1C—USE OF FIG. 1 TO FIND DESIGN PANEL OUTPUT (STEP 8)

Step 4—Downward Heat Flow Estimate: (1) Follow the procedure for slab-on-grade construction, Step 4, part 1. (2) Estimate the heat flow downward as follows: (*a*) If the underside of the concrete slab is exposed to form a ceiling, find the heat output downward (q_d) from Fig. 1 of Reference 1, using the slab surface temperature (t_s) found in Step 3 as the panel surface temperature (t_p), and the air

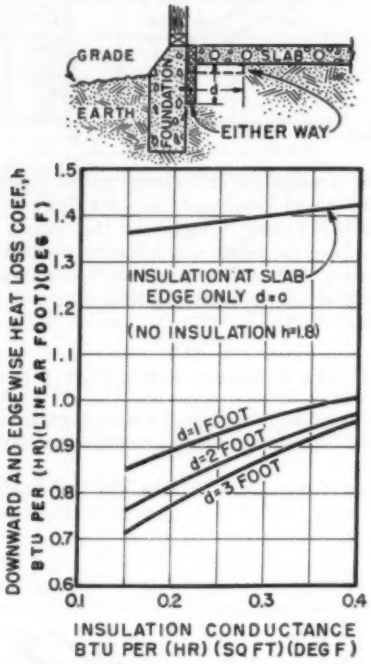

FIG. 2—DOWNWARD AND EDGEWISE HEAT LOSS COEFFICIENT FOR CONCRETE FLOOR SLABS ON GROUND

temperature of the space below. (*b*) If the underside of the concrete slab is not exposed, use the equation:

$$q_d = C\,(t_s - t_b)$$

Step 5—Upward and Downward Panel Resistance: Follow the procedure for slab-on-grade construction, Step 5, using Table 2 to find both resistances (r_{us} and r_{ds}).

Step 6—Mean Water Temperature and Downward Heat Flow: (1) To determine the mean water temperature, follow the procedure for slab-on-grade construction, Step 6. (2) Use Fig. 1 of Reference 1 to find the downward heat flow (q_d). Add to the slab resistance to downward heat flow, (r_{ds}), the resistance to heat flow, (r_{de}), of any material between the underside of the slab and the ceiling surface, to find the total resistance to downward heat flow (r_d) to be used in Fig. 1 of Refer-

ence 1. The water temperature (t_{mw}) found above and the air temperature (t_b) of the space below the ceiling are the other two factors to be used. If the heat flow downward (q_d) differs appreciably from the estimate made in Step 4, repeat Steps 5 and 6 using the calculated value.

Step 7—Design Mean Water Temperature: Follow the procedure for slab-on-grade construction, Step 7.

NOMENCLATURE

A_p = panel area, square feet.
C = coefficient of heat transfer, Btu per (hour) (square foot) (Fahrenheit degree) taken from lower surface of concrete slab to air below the panel (temperature, t_b).
h = downward and edgewise heat loss coefficient, Btu per (hour) (linear foot of exposed slab perimeter) (Fahrenheit degree) from concrete surface to outdoor air.
P = length of exposed edge of slab, feet.
q_d = downward heat flow from panel, Btu per (hour) (square foot).
q_{de} = apportioned downward and edgewise heat flow from panel, Btu per (hour) (square foot).
q_u = panel output (upward heat flow), Btu per (hour) (square foot).
r_d = total thermal resistance of panel to downward heat flow (Fahrenheit degrees) (hour) (square foot) per Btu.
r_{de} = thermal resistance of material between the underside of the concrete slab and the ceiling surface below (Fahrenheit degrees) (hour) (square foot) per Btu.
r_{ds} = thermal resistance of bare concrete slab to downward heat flow, (Fahrenheit degrees) (hour) (square foot) per Btu.
r_u = total thermal resistance of panel to upward heat flow (Fahrenheit degrees) (hour) (square foot) per Btu.
r_{ue} = thermal resistance of floor covering (Fahrenheit degrees) (hour) (square foot) per Btu.
r_{us} = thermal resistance of bare concrete slab to upward heat flow (Fahrenheit degrees) (hour) (square foot) per Btu.
t_a = design room air temperature, Fahrenheit.
t_{ao} = outside design air temperature, Fahrenheit.
t_b = air temperature below panel at point to which C is taken, Fahrenheit.
t_{mw} = mean water temperature, Fahrenheit.
t_{dmw} = design mean water temperature (selected for each zone) Fahrenheit.
t_p = panel surface temperature (exposed surface) Fahrenheit.
t_s = surface temperature of top of concrete slab, Fahrenheit.

Step 8—Design Panel Output: Follow the procedure for slab-on-grade construction, Step 8.

Step 9—Design Panel Area: Follow the procedure for slab-on-grade construction, Step 9.

Step 10—Total Panel Output: Follow the procedure for slab-on-grade construction, Step 10, substituting heat flow downward (q_d) for the apportioned downward and edgewise heat flow, (q_{de}).

Step 11—Fluid Circuit: Follow the procedure for slab-on-grade construction, Step 11.

Step 12—Boiler Size: Follow the procedure for slab-on-grade construction, Step 12.

APPENDIX

All the figures and tables included in the design procedure are based on data developed at the ASHAE Research Laboratory, unless otherwise noted. The methods by which the figures and tables were derived are identified in this Appendix.

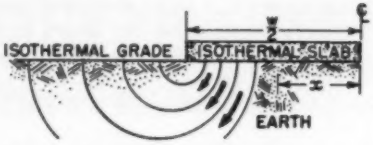

FIG. A-1—HEAT FLOW PATTERN BETWEEN FLOOR SLAB AND GRADE

FIG. 1—FLOOR PANEL DESIGN GRAPH

Fig. 1 and Fig. 1 of Reference 1 are similar in form and use. Both were plotted from points calculated as outlined in the Appendix to Reference 1. The equations used in the calculations for Fig. 1 were:

$$t_p = [(t_a + \Delta t) - AUST]/0.32 + AUST \quad \text{(A-1)}$$

(Reference 2, Fig. 7 and 10)

$$q_{ru} = 1.52 \times 10^{-9} [(t_p + 460)^4 - (AUST + 460)^4] \quad \text{(A-2)}$$

(Reference 3)

$$q_{cu} = 0.31 [t_p - (t_a + \Delta t)]^{1.31} + \Delta q_{cu} \quad \text{(A-3)}$$

(Reference 3 and Reference 2, Fig. 4.)

Some panel performance data calculated by means of Equations A-1, A-2, and A-3 are given in Table A-1.

The family of curves relating floor panel output (q_u) to mean water temperature (t_{mw}) were plotted from the equation:

$$t_{mw} = t_p + r_u q_u \quad \text{(A-4)}$$

The correcting section of the abscissa was constructed as described in the Appendix to Reference 1.

Test conditions were as given in the Appendix to Reference 1 and the comments contained therein also apply to Fig. 1 of this paper. Additional References are 4, 5, 6 and 7.

FIG. 2—DOWNWARD AND EDGEWISE HEAT LOSS COEFFICIENT FOR FLOOR SLABS ON GRADE

The heat loss coefficient given by Fig. 2 was calculated to combine into a single coefficient the downward heat loss and the edgewise heat loss of a floor slab on grade.

TABLE A-1—SOME RESULTS OF PANEL PERFORMANCE CALCULATIONS

t_a	72	72	72	72
$AUST$	71	72	73	74
Δt	3.6	3.6	3.7	3.7
t_p	85.4	83.3	81.5	79.3
q_{ru}	13.7	10.6	8.0	4.9
q'_{cu}	6.2	4.5	3.2	1.7
Δq_{cu}	6.9	6.9	6.8	6.7
q_u	26.8	22.0	18.0	13.3

The coefficient is based on the exposed perimeter of the slab and the difference in temperature between the slab and the outside air.

The downward heat loss from the panel shown in Fig. 2 was approximated by calculating the heat flow from the slab shown in Fig. A-1, modifying the results to allow for the difference in arrangement between the slabs in the 2 figures. The procedure used was as follows:

The slab shown in Fig. A-1 is infinitely long and is on ground which is of infinite width. The heat flow through the ground from the slab per degree temperature difference between the 2 isothermal surfaces, is:

$$h_s'' = \frac{k_s}{\pi} \ell n \sqrt{\frac{\left(x - \frac{w}{2}\right)^2}{\left(x + \frac{w}{2}\right)^2}} \quad \text{------(A-5)}$$

This equation was derived in the ASHAE Laboratory, using the Schwarz-Christoffel transformation to change a simple straight line heat flow pattern to that illustrated in Fig. A-1. It shows that the heat flow is infinite at the exposed edge ($x = w/2$). In practice the heat flow is not infinite because there is usually at least 8 in. of concrete between the edge of the slab and the exposed grade. The assumption was made that the heat flow at any point would be no greater than if 8 in. of concrete (conductivity 1.5) were the only thermal resistance. That is, the rate of heat loss.

$$\frac{dh_s''}{dx} = \frac{2k_s}{\pi} \frac{\frac{w}{2}}{\left(x - \frac{w}{2}\right)\left(x + \frac{w}{2}\right)} \leqq 1.5 \quad \text{----------(A-6)}$$

The heat flow was calculated separately for each of the first 3 ft and for the balance of the slab. The apparent heat flow resistance for each of the first 3 ft was then calculated from the heat flow. The resistance of the first foot of insulation was assumed to be in series with the calculated resistance of the first foot of slab, the second foot of insulation with the second foot of slab, the third foot of insulation with the third foot of slab. The total resistance of each foot was used to calculate the heat flow from the slab.

A factor to convert the heat flow (h'_s) from an infinitely long slab to the heat flow (h_s) from a slab of given length was developed by Macey, Reference 8. The adaptation of the factor which was used was:

$$h_s/h'_s = (0.6w + l)/(w + l) \quad \text{. . . . . (A-7)}$$

Edgewise heat loss was calculated from:

$$h_e = \frac{\frac{L}{12}}{\frac{1}{f_o} + \frac{X_1}{k_1} + \left(\frac{X_2}{k_2} \text{ or } \frac{1}{C_i}\right)} \quad \text{------(A-8)}$$

The combined coefficient is the sum of the downward and edgewise coefficients:

$$h = h_s + h_e \quad \text{. . . . . (A-9)}$$

Fig. 2 was plotted from points determined by calculating the losses from a 24 ft x 36 ft slab, 4 in. thick. Conductivity of concrete was assumed to be 1 Btu per (hr)(sq ft)

(F per foot) and conductivity of soil 10/12 Btu per (hr)(sq ft)(F per foot). Equation A-6 was used to determine the point at which $dh''_s/dx = 1.5$. It was found to be at $x = 11.8$ ft and $dh''_s/dx = 1.5$ was used from $x = 11.8$ ft to $x = 12.0$ ft. Other values are tabulated in Table A-2.

Corrected for 24 ft x 36 ft slab,

$$h_s = 1.57\ [(0.6 \times 24) + 36]/(24 + 36) = 1.57 \times 0.84 = 1.32$$

If insulation having a conductance $C_1 = 0.4$ is used under the first 3 ft of the slab, the heat loss is reduced to

$$h'_s = [1/(1.35 + 1/0.4)] + [1/(5.26 + 1/0.4)] + [1/(8.33 + 1/0.4)] + 0.52 = 1.00$$

Corrected for 24 ft x 36 ft slab,

$$h_s = (1.00 \times 0.84) = 0.84$$

TABLE A-2—DATA USED IN DETERMINING FIG. 2

LOSS AREA	FROM x_1 TO x_2	HEAT FLOW (h''_s TO x_1 − h''_s TO x_2)		RESISTANCE (h''_s TO x_1 − h''_s TO x_2)$^{-1}$
Part 1st ft	12.0–11.8	(0.2 x 1.50)	0.30	
Part 1st ft	11.8–11.0	(1.27–0.83)	0.44	
Total 1st ft	12.0–11.0		0.74	1.35
2nd ft	11.0–10.0	(0.83–0.64)	0.19	5.26
3rd ft	10.0– 9.0	(0.64–0.52)	0.12	8.33
Bal. of slab	9.0– 0.0	0.52	0.52	
Total slab	12.0– 0.0		1.57	

Slab edge loss for a 4 in. slab and a 6 in. thick footing without insulation is

$$h_s = (4/12)/(1/6 + 6/12) = 0.50$$

and with insulation, $C_1 = 0.4$

$$h_s = (4/12)/(1/6 + 6/12) + (1/0.4) = 0.11$$

The combined downward and edgewise heat loss coefficient is thus:

$h = 1.32 + 0.50 = 1.82$ No insulation.
$h = 1.32 + 0.11 = 1.43$ Insulated at edge only, $C_1 = 0.4$.
$h = 0.84 + 0.11 = 0.95$ Insulated at edge and for 3 ft under slab, $C_1 = 0.4$.

The heat loss coefficients given by Fig. 2 should be considered as approximations because of the simplifying assumptions made in their calculation and because of the differences in construction and thermal conductivities likely to be met in practice. In some cases, it may be advisable to use Equations A-5, A-6, A-7, and A-8 to calculate the heat loss, adjusting them to construction involved. In most cases, however, the value of the coefficient given in Fig. 2 should be sufficiently accurate for practical heating calculations involving either heated or unheated floor slabs.

TABLE 1—THERMAL RESISTANCE OF FLOOR COVERINGS

The thermal resistances tabulated were calculated from data reported in References 6 and 9.

TABLE 2—THERMAL RESISTANCE OF BARE CONCRETE FLOOR PANELS

That data for this table were obtained in exactly the same way as were the data for Table 2, Reference 1. See also Reference 10.

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8. Heat Loss Through a Solid Floor, by H. H. Macey (*Journal of the Institute of Fuel*, 1949, 22 (128), p. 369).

9. Performance of Covered Hot Water Floor Panels, Part I—Thermal Characteristics, by E. L. Sartain and W. S. Harris, (ASHAE TRANSACTIONS, Vol. 62, 1956, p. 55).

10. ASHVE RESEARCH REPORT No. 1388—Heat Flow Analysis in Panel Heating or Cooling Sections, Case I—Uniformly Spaced Pipes Buried Within a Solid Slab, by L. E. Hulbert, H. B. Nottage, and C. V. Franks (ASHVE TRANSACTIONS, Vol. 56, 1950, p. 189).

NOMENCLATURE IN APPENDIX

The following terms used in the Appendix are not defined in the paper:

$AUST$ = area-weighted average temperature of unheated surfaces in room, Fahrenheit.

x = dimension used in Equations A-5 and A-6, (see Fig. A-1), (feet).

C_i = conductance of insulating material at slab edge, Btu per (hour) (square foot) (Fahrenheit degree).

f_o = outside film conductance, Btu per (hour) (square foot) (Fahrenheit degree).

h_e = heat loss from edge of slab, Btu per (hour) (foot) (Fahrenheit degree).

l = slab length, (feet).

k_s = conductivity of soil, Btu per (hour) (square foot) (Fahrenheit degree per foot).

k_1, k_2 = conductivity of material, Btu per (hour) (square foot) (Fahrenheit degree per inch).

L = thickness of slab, (inches).

q_{cu} = convection component of panel output (upward heat flow), Btu per (hour) (square foot).

q'_{cu} = convection component of panel output (upward heat flow), no infiltration, Btu per (hour) (square foot).

Δq_{cu} = increase in convection component of panel output (upward heat flow) caused by infiltration, Btu per (hour) (square foot).

q_{ru} = radiation component of panel output (upward heat flow), Btu per (hour) (square foot).

h'_e = heat loss through earth per foot of exposed edge of floor slab of infinite length (see Equation A-7), Btu per (hour) (foot) (Fahrenheit degree).

h''_e = heat loss through earth for width x, per foot of exposed edge of floor slab of infinite length (see Fig. A-1 and Equation A-5), Btu per (hour) (foot) (Fahrenheit degree).

$\frac{dh''_e}{dx}$ = rate of heat loss through earth at distance x, per foot of exposed edge of floor slab of infinite length (see Fig. A-1 and Equation A-6), Btu per (hour) (square foot) (Fahrenheit degree).

h_e = heat loss through earth per foot of exposed floor slab edge of length l (see Equation A-7), Btu per (hour) (foot) (Fahrenheit degree).

Δt = decrease in room air temperature caused by infiltration (see Fig. 10, Reference 2), Fahrenheit.

w = slab width, feet.

X_1, X_2 = thickness of materials, inches.

DISCUSSION

E. L. SARTAIN, Racine, Wisc. (WRITTEN): Much time and effort was apparently put into the development of this design procedure. The authors are to be commended for their work of resolving a very complicated collection of data into an excellent and workable simplified design procedure for floor panels.

Two points have been mentioned which I would like to discuss. The first of these has to do with the range of application of this data. The authors point out that this design procedure is *restricted to situations in which the area-weighted average temperature of the walls, the ceiling, and glass does not differ greatly from room air temperature.* Under the section *Range of Application* the authors further point out that the design procedure applies to *any type of wall construction and any amount of glass area,* and further that both of these have an effect on comfort. It would seem that any change in environmental conditions would necessarily result in changes in surface and air temperature equilibrium, and thus would effect heat flow. Measurements made at the University of Illinois Floor Slab Laboratory, in rooms having ordinary construction, showed a difference between room-air temperature and AUST of at least 4 deg F for −10 F outdoor temperature. Since this 4 deg F difference in room-air temperature and AUST was obtained in rooms with somewhat conservative exposure, it is quite possible that a room having very little infiltration and large amounts of glass area would exhibit a higher difference in room-air temperature and AUST. Such being the case, the writer would like to know what the limiting value of room-air AUST temperature difference is beyond which this design procedure should not be used.

The second point for consideration has to do with boiler size. Under step 12 in this design procedure, the authors recommend that a boiler be sized according to the method contained in THE GUIDE. A piping and pick-up factor of 33 percent is allowed by the *Institute of Boiler and Radiator Manufacturers* for residential boiler sizes. It was shown in a previous paper (*Performance of Covered Hot Water Floor Panels—Part I* by Sartain and Harris) that with heavy carpets and pads over floor panels, considerable increase in boiler size was required for additional pick-up allowance. This additional heat input was necessary because of increased heat storage due to the required increase in temperature of the system including that of the large mass of concrete. With insufficient boiler capacity, it was found that with rapid outdoor temperature drops, room-air temperature was lowered to a value which could have resulted in discomfort. For these reasons the use of a standard pick-up factor is questioned when applied to a boiler serving panels covered by heavy carpets and pads.

F. J. LINSENMEYER, Detroit, Mich. (WRITTEN): I wish to compliment the Society, the members of the Technical Advisory Committee and the Research Staff for such a thorough yet simple standard design procedure for floor panels. It complements the standard for ceilings read before this Society last year.

I should like to point out that the calculated heat loss of room A in this paper is 3300 Btuh, whereas the loss for a similar room in the paper on ceiling heating is 6300. It appears to me that a room with 2 exposures would require very low U values to make it applicable—and still lower if we chose a minus 30 F design. Since floor radiant is frequently designed for institutional buildings with single glass and un-insulated walls, a more stringent limitation should be put on the outside design temperature and infiltration.

The ceiling radiant heating for the same rooms (room A 6300) requires water at 131 F versus 116.5 in the floor radiant (room A 3300). The coiled area in the ceiling radiant was 451 square feet, while in the floor it was 542.

Another comparison may be made between the 2 forms of heating. If the ceiling design loss is taken the same as the floor, the gross input to the space for the ceiling is 16800 Btuh versus 20497 for the floor or 18 percent less heat required from the boiler. This advantage would be furthered if edge insulation were used in the ceiling radiant job.

I have noted some errors in symbols but assume the proof reader has made those corrections. I note that heavy carpet is used without under-padding. I believe most carpet is laid with padding.

I do not see in Step 4 Intermediate slab where credit is given to the downward heat flow in the space heating to make up for the heat loss.

J. L. BRADFIELD, Indianapolis, Ind.: No doubt many of you were at our Dallas meeting in 1950 when Herrington and Lorenzi of the Pierce Foundation Laboratories at New Haven reported on extensive tests they made, comparing the effect on the human body, of radiation from the ceiling and from the floor. They used Yale students as *guinea pigs* and definitely established the fact that a floor surface temperature above 75 F. is uncomfortable and physiologically wrong.

There are few heating areas in this country where the heat loss of a room can be balanced with a 75 F floor panel.

Since this information is available to our members, why should our laboratory staff waste time and money on further tests of floor panels?

GEORGE APPLEGATE, JR., Trowbridge, England: Being connected with a firm experimenting on floor heating, we in England are very conscious of fuel costs. We find that floor heating through the period of heating up is a definite cost which we cannot now stand and I would like to see if you have any data on running costs with regard to downward loss of heat which is useless also any known method of quick response to sudden climatic temperature changes.

E. L. SARTAIN, Racine, Wisc.: In regard to questions raised by Mr. Applegate concerning operating costs for floor panels *vs.* ceiling panels, I would like to make some comment based on investigations of both floor and ceiling panel performance at the University of Illinois. Here, both a floor and ceiling panel were operated independently in the same test room during consecutive heating seasons. It was found that, in general, the cost of operating the 2 different systems was greater for the ceiling panel. However, I would like to qualify this statement and point out that comparative heat losses depended entirely on the condition of the space above the ceiling panel and the method of installing tubes.

With a normally ventilated attic and a conventional ceiling panel having coils immersed in plaster, and insulated with medium thick rock wool batts, it was found that when the coil was located above metal lath with no back plastering the heat loss to the attic became rather excessive. With the coil located below the lath, the heat loss to the attic was not as great as with the coil above the lath. However, in both cases, the

heat losses to the attic exceeded those obtained with a floor panel to the extent that the annual fuel bill for a heating season was increased.

This information and test data are being prepared for publication in an Engineering Experimental Station Bulletin to be published by the University of Illinois.

P. B. GORDON, New York, N. Y.: I would like to provide my own comments. There are no set rules or no set pattern for establishing what is the best for any given installation. One would have to evaluate the several systems. And there might be reasons why one would take a deliberate loss yet gain other advantages—such as simplicity. The designer could be in a position of not being able to install ceiling panels, and you may therefore want floor panels. There is no set pattern for establishing in advance that which is the best. One must consider the advantages and disadvantages of both arrangements.

AUTHOR'S CLOSURE (L. F. Schutrum, Cleveland, Ohio): I wish to thank Mr. Linsenmeyer for his comparison between a floor panel heating system and a ceiling panel heating system. In regard to Step 4, intermediate slab, the downward heat flow from the floor panel to the space below must be considered as a gain of heat for this space. This heat gain is part of the heat loss calculation Step 1, and by the method of trial and error, a proper heat balance can be made. The quantity of heat flow downward is obtained from Fig. 1 of Reference 1 for a concrete floor panel which is exposed on the bottom to form a ceiling.

Mr. Sartain asked for clarification of the range of application of the design since it is stated in the paper that the design procedure be restricted to situations in which the area-weighted average temperatures of the walls, the ceiling, and glass does not differ greatly from the room air temperature. He mentioned that in his tests, this difference was of the order of 4 deg F. Actually, in the experimentation at the Laboratory, the temperature difference between the room air and the area-weighted average temperature of the floor, four walls and windows, was up to about 18 deg F. Even with this extreme condition, the design graph, Fig. 1, was found applicable.

Mr. Sartain questioned the piping and pickup factor of 33 percent which is used in Step 12 to size the boiler. His criticism is probably correct in that the normal pick-up factor may not be sufficient to compensate for the large thermal storage encountered in concrete floor panels with insulating floor coverings such as a rug and mat combination. The work of Mr. Sartain and Professor Harris in their series of papers on floor panel heating systems has contributed much to this thermal design procedure.

Mr. Bradfield raised the question as to the maximum floor surface temperature which can be used without discomfort. We do not as yet have the complete answer to this problem, however, one of the research programs which will be carried on at the Society's Laboratories in Cleveland is the study of human comfort under radiant conditions. The surface temperature of the floor of the test room will be controlled from 40 F to 100 F and a subjective study will be made.

In closing, I would like to thank the members of the Subcommittee for their contributions to this paper.

1601

No. 1601

OUTLET CHARACTERISTICS THAT AFFECT THE DOWNTHROW OF HEATED AIR JETS

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, in cooperation with the Engineering Experiment Station of Kansas State College.

By LINN HELANDER*, S. M. YEN**, AND WILSON TRIPP***, MANHATTAN, KANS.

DATA OBTAINED from research pursued at Kansas State College under the guidance of the TAC on Air Distribution and having guidance value for designers and application engineers are presented, herein, on the actual thrust (projectile force) and the relative thrust, β_2, developed by (1) a 4-in. *ASME* nozzle with a remote fan and (2) a unit heater with the fan immediately ahead of the outlet. The unit heater was equipped in one case with a deep cylindrical outlet, in the other, with a shallow diffuser. Data are presented also on the change in the actual and relative thrusts produced by:

1. First varying the vane angle, then removing the vanes with which the shallow diffuser was originally equipped.
2. Inserting short cores into each of the outlets with the downstream face of the core flush with the face of the outlet.
3. Placing a plate or shield in the jet at different distances downstream from the outlet.

The relative thrust, β_2, is defined as the ratio of the actual thrust developed by an outlet to the apparent thrust calculated for zero gage-static pressure and a uniform velocity, Q_0/A_0, across the face of the outlet. In the present instance, β_2 was evaluated by directly measuring the reactive force of an unheated, horizontal jet. For the *ASME* outlet, the values of β_2 so obtained are directly applicable when the outlet is used to produce a downwardly projected jet of heated air. For the other outlets so used, some correction to the values of β_2 obtained may prove necessary.

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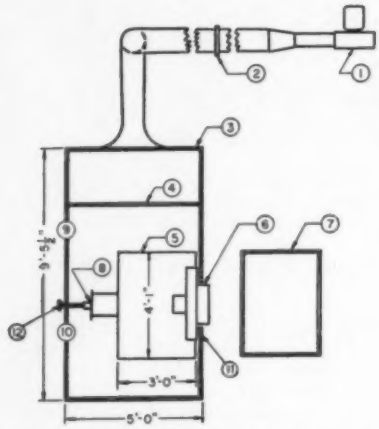
Presented at the 63rd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Chicago, February 1957.

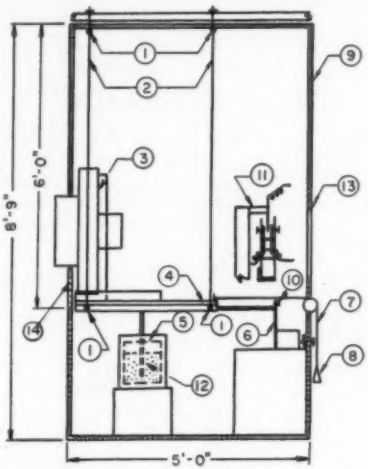
Tests are planned to determine values of β_2 for heated, horizontally projected jets, and, to the extent observable, the effect on β_2 produced by using the equipment tested to develop downwardly projected heated jets. In the absence of such data, values of β_2 presented herein will be regarded as indicative of the values of β_2 for the equipment tested when this equipment is used to produce heated jets discharged vertically downward. This would seem to be justified, so long as this use of the equipment does not alter the delivery of the fan in cfm at its established speed.

An idealized theory of downwardly projected jets is reviewed briefly to show that if an outlet of given diameter is modified to increase the product, $\beta_2 \overline{B}_0$, in which



Reference numbers: (1) blower, (2) metering nozzle, (3) housing, (4) baffle, (5) swing platform, (6) unit heater mounted on swing platform, (7) instrument carriage, (8) thrust measuring device, (9) door, (10) window, (11) seal, and (12) means for restoring plumb position of swing platform. The door and window were closed and atmospheric pressure was maintained in the housing by means of the external fan (1).

FIG. 1—DIMENSIONS AND SCHEMATIC ARRANGEMENT OF EQUIPMENT (PLAN VIEW) FOR MEASURING OUTLET THRUST OF UNIT HEATERS (TO SCALE)



Reference numbers: (1) ball bearings, (2) support rods, (3) unit heater, (4) swing platform, (5) vibration damper, (6) thrust-measuring beam, (7) string, and (8) calibration weights (not present when measuring thrust), (9) housing, (10) position-indicating switch, (11) enlarged top view of position-indicating switch, (12) vibration—damper oil, (13) window, and (14) seal.

FIG. 2—DIMENSIONS AND SCHEMATIC ARRANGEMENT OF EQUIPMENT, (SIDE ELEVATION) FOR MEASURING OUTLET THRUST OF UNIT HEATERS (TO SCALE)

$\overline{B}_0$ is the buoyancy number (see Nomenclature), and offsetting forces are not developed at the boundaries of the space heated, the increase in $\beta_2 \overline{B}_0$ will produce one or, to a lesser extent, both of the following effects:

1. An increase in the downthrow expressed in outlet diameters.
2. A decrease in the magnitude of the radial temperature gradient and thus a radial expansion of the sensible thermal boundaries of the jet.

The same effects will be produced by an increase in the outlet downthrust. In either case, the effect or effects produced will depend upon the design of the outlet and the modifications made to it.

β_2 is shown to be one component of the factor, f, in the downthrow equation:[3]

$$L_{max}/D_o = 1.7 f \bar{B}_o^{\frac{1}{2}}$$

Thus

$$1.7f = [K_T(1+p)\beta_2]^{\frac{1}{2}}\dagger$$

where

K_T = a factor dependent on the pattern of temperature distribution in the room heated (K_T decreases as the magnitude of the radial temperature gradient decreases).

p = the increment of vertical boundary forces (positive downward) due to dynamic effects induced by the jet, expressed as a fraction of the outlet thrust.

β_2 = principally a function of the velocity profile and the average gage, static pressure at the face of the outlet. (Negative static pressures at the outlet decrease β_2; non-uniformities of local velocities at the outlet increase β_2).

Whirl at the outlet significantly reduced the magnitude of β_2. Thus the average value of β_2 for the shallow diffuser and the cylindrical outlet, both without vanes or cores, and both served directly by the unit heater fan was 0.78, whereas β_2 for the whirl-free *ASME* outlet was 1.03. For the unit-heater, shallow-diffuser combination, vanes introduced into the diffuser and set vertical to the face of the outlet increased the magnitude of β_2 20 percent.

When cores were employed, an increase in the size of the core increased β_2 but reduced the rate of air delivery (the fans were operated at constant speed). Consequently, the thrust developed by an outlet first increased, then decreased as the core size was increased. The maximum thrust developed by the *ASME* nozzle and the vane-free shallow diffuser was realized when the ratio of the core area to the outlet area was between 0.3 and 0.5.‡ The corresponding rate of air delivery was 10 to 20 percent below the rate developed without cores. For the shallow diffuser with its original guide vanes, the maximum thrust and the maximum rate of air delivery were realized when the guide vanes were set perpendicular to the face of the outlet. Again the fan was run at one speed.

A 2-in. core inserted into the *ASME* nozzle had no significant effect on either the velocity or the shape of the velocity profile of the jet beyond a distance equal to 3 outlet diameters downstream from the outlet, although the volume rate of flow at the outlet was reduced 13 percent.

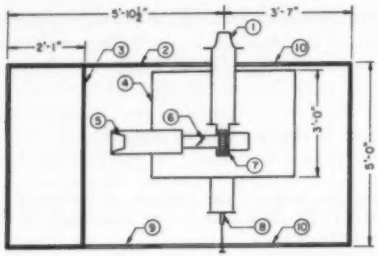
For an unrestricted flow of 4 cfs (240 cfm), the shield inserted into the jet from the 4-in. *ASME* nozzle had no significant effect on outlet conditions when placed more than one outlet diameter downstream from the outlet.

EQUIPMENT AND PROCEDURE

The thrust developed by an outlet was measured by mounting the outlet on a swing platform and measuring the reactive force of the jet produced by the outlet. A photograph of the test station is shown in Fig. A-1 of the Appendix. Plan and elevation views are shown schematically in Figs. 1 and 2. Fig A-2 of the Ap-

† Some other arrangement of empirical factors may prove preferable from the viewpoint of correlating experimental data. The arrangement adopted is convenient for present purposes.

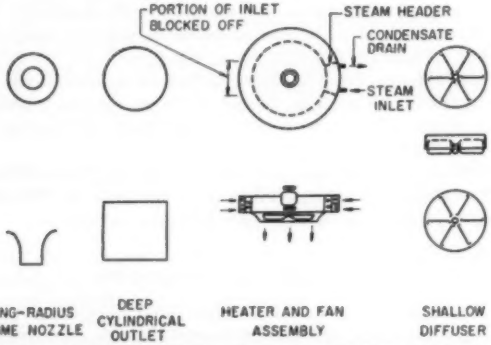
‡ For an outlet of fixed diameter, the maximum thrust occurs where $\beta_2 Q_o^2$ is a maximum.



The door (9) and the windows (10) were open so as to maintain atmospheric pressure in the housing. Reference numbers: (1) *ASME* nozzle, (2) housing, (3) baffle, (4) swing platform, (5) metering nozzle, (6) damper, (7) blower, (8) thrust-measuring device, (9) open door, (10) open windows.

FIG. 3—DIMENSIONS AND SCHEMATIC ARRANGEMENT OF EQUIPMENT (PLAN VIEW) FOR MEASURING OUTLET THRUST OF *ASME* NOZZLE (TO SCALE)

pendix is a photograph of the thrust-measuring device. This consisted of SR-4 type A5 strain gages mounted on a 6 x ½ x 1/16-in. vertical cantilever beam which was pressed against the swing platform (Fig. 2) so as to counterbalance the reaction of the jet and maintain the swing platform in a plumb position. Two lights connected to a position-indicating switch flickered at equal rates when the platform was in a plumb position (Fig. 2). The thrust-measuring device was calibrated by means of weights (Fig. 2). It was sensitive to a force of one gram applied to the swing platform as a simulated thrust. The manner in which the unit heater was mounted on the platform is shown in Fig. 2. Fig. 3 shows the manner in which the

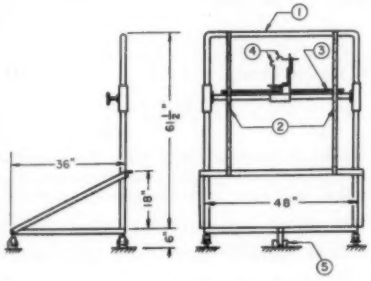


Dimensions of outlets: *ASME* nozzle, outlet diameter 4 in.; cylindrical outlet, diameter 17¼ in., length 13 5/16 in.; shallow diffuser, diameter, 17½ in.

FIG. 4—SCHEMATIC DRAWINGS OF LONG-RADIUS *ASME* NOZZLE, HEATER AND FAN ASSEMBLY, AND OUTLETS

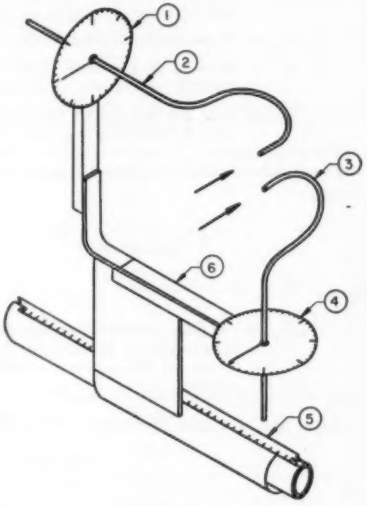
ASME nozzle was mounted. Air at atmospheric pressure was delivered to the equipment on the swing platform in a direction at right angles to the axis of the outlet. The rate of flow of air to the outlets was regulated by dampers and measured by measuring nozzles (Figs. 1 and 3). The fan that served the *ASME* nozzle and the unit heater fan were each operated at a single speed.

Fig. 4 shows the types of outlets employed. Fig. 5 shows the means employed for making velocity traverses at the outlet and in the jet downstream from the outlet. Various sizes and shapes of cores (Tables 1 and 2) were employed in the outlets without vanes. These cores were held in place by an external bracket attached to the outlet. Impact tubes used for determining the direction of flow of



Reference numbers: (1) carriage frame, (2) vertical scale, (3) horizontal scale, (4) impact tubes, (5) guide rail parallel to axis of outlet. The instrument carriage could be moved horizontally, *i.e.*, in the direction of flow. The impact tubes (4) could be moved in a vertical plane to any point within the frame (1) of the carriage.

FIG. 5—DIMENSIONS AND SCHEMATIC ARRANGEMENT OF INSTRUMENT CARRIAGE



Reference numbers: (1) vertical circular scale, (2) impact tube for velocity component in vertical plane, (3) impact tube for velocity component in horizontal plane, (4) horizontal circular scale, (5) horizontal scale, (6) frame.

FIG. 6—SCHEMATIC DRAWING OF AIR-FLOW DIRECTION MEASURING DEVICE

air at the face of the unit heater outlet are shown in Fig. 6. Velocities and static pressures were measured by means of impact and static-pressure tubes.

To check the accuracy of the thrust-measuring device, the thrust developed by the *ASME* nozzle was evaluated from the velocity profile and the result was compared with the measured thrust. The agreement was good.

IDEALIZED THEORY

That the downthrust, τ_0, and the product, $\beta_2\overline{B}_0$, have the properties attributed to them in the introduction can be established rather simply by a brief review of an

idealized theory of downwardly projected heated jets. To do this, the following assumptions will be employed:

1. The outlet will be located at the ceiling of the room heated.
2. The resultant of all vertical forces, including the outlet downthrust, acting within and at the boundaries of the room heated by the jet will be zero.
3. The resultant of the buoyancy forces acting within the room can be evaluated as

$$(g/g_c)(L_{max})(\rho_a - \rho_{av})A_s$$

where

L_{max} = the throw, defined as the distance from the outlet to the level below which the jet produces no further sensible effect (the air below this level will be treated as still air at a uniform temperature, T_a).

A_s = cross-sectional area of the room heated.

ρ_a = density of a room-height column of air located where the air may be treated as still air at a uniform temperature, T_a.

ρ_{av} = average density of the volume of room air (average temperature, T_{av}) which extends down from the level of the outlet, L_{max} feet.

These assumptions lead to Equation 1 wherein the left-hand side is the buoyancy force:

$$\frac{g}{g_c}(L_{max} A_s \rho_{av}) \frac{T_{av} - T_a}{T_a} = (\text{Outlet Thrust} + \Delta F_b) \quad . \quad . \quad . \quad . \quad (1)$$

where

ΔF_b = vertical resultant of dynamic forces at boundaries of room due to dynamic effects induced by the jet (positive acting downward on room air). Data on ΔF_b are not available; ΔF_b will vary with the geometry of the room.

Equation 1 takes the form of the downthrow equation previously presented[3]:

$$L_{max}/D_0 = 1.7 f \bar{B}_0^{1/3}$$

if

$$(L_{max} A_s \rho_{av}) \left(\frac{(T_{av} - T_a)}{T_a} \right) = \pi/4 \, L^2_{max} D_0 (\bar{T}_0 - T_a) \frac{\rho_0}{K_T T_a} \quad . \quad . \quad . \quad (2)^x$$

$$\Delta F_b = (p)(\text{Outlet Thrust}) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)\square$$

$$(\text{Outlet Thrust} + \Delta F_b = \pi/4 \, (1 + p) \, (\beta_2 \bar{U}_0^2 \rho_0 D_0^2/g_c) \quad . \quad . \quad . \quad . \quad (4)$$

Thus, using Equations 2 and 3 in Equation 1:

$$\pi/4 \frac{g}{g_c} (L^2_{max} D_0)(\bar{T}_0 - T_a) \frac{\rho_0}{K_T T_a} = (1 + p)(\text{Outlet Thrust}) \quad . \quad . \quad . \quad (5)$$

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[x] The form of Equation 2 is derivable analytically if the temperature point source is assumed to be at the outlet and if, *first*, $(T_0 - T_a)/(\bar{T}_0 - T_a) = L_p/L$ and, *second*,

$$(T - T_a)/(T_c - T_a) = e^{(-K'_T r^2/L^2)}$$

then $K_T = K'_T/(2L_p/D_0)$.

□ Even with a fixed geometry of room, there is no present reason for expecting p to be a constant.

TABLE 1—OUTLET FLOW DATA FOR UNIT HEATER WITH (1) A CYLINDRICAL OUTLET AND (2) A SHALLOW DIFFUSER, BOTH WITH AND WITHOUT CORES AND WITHOUT VANES

OUTLET	CORE USED (6 IN. DIAM.)	$\frac{Q_0}{Q_{0,\text{ no core}}}$	$\frac{\tau_0}{\tau_{0,\text{ no core}}}$	β_2	C''_p	$\Delta\beta''_{2p}$	$\frac{(\Sigma U^2)/N}{[(\Sigma U)/N]^2}$
Cylindrical Outlet	Hemisphere	0.97	1.04	0.85	−0.33	0.053	1.59
Cylindrical Outlet	Disc	0.97	1.03	0.85	−0.34	0.054	1.61
Cylindrical Outlet	No Core	1.00	1.00	0.77	—	—	—
Shallow Diffuser[a]	Hemisphere	0.97	1.08	0.89	−0.25	0.041	1.92
Shallow Diffuser[a]	Disc	0.98	1.07	0.87	−0.28	0.044	1.84
Shallow Diffuser[a]	No Core	1.00	1.00	0.79	—	—	1.53

$$\frac{[(Q_0)\text{ shallow diffuser}^a]_{\text{no core}}}{[(Q_0)\text{ cylindrical outlet}]_{\text{no core}}} = 1.03 \; ; \; \frac{[(\tau_0)\text{ shallow diffuser}^a]_{\text{no core}}}{[(\tau_0)\text{ cylindrical outlet}]_{\text{no core}}} = 1.11.$$

[a] Without guide vanes.

$C''_p = -(P_a - \overline{P''}_0)/[Q_0/(A_0 - A_c)]^2(\rho_0/g_c)$; $\Delta\beta''_2 = -C''_p[A_0A_c/(A_0 - A_c)^2]$ = relative drag at downstream face of core; $[(\Sigma U^2/N)]/[(\Sigma U/N)]^2$ = an index such that the larger its value the more non-uniform the velocity at the outlet;

$[(Q_0)\text{ cylindrical outlet}]_{\text{no core}} = 27.5$ cfs (1650 cfm); $[(\tau_0)\text{ cylindrical outlet}]_{\text{no core}} = 0.60$ lb.

Using Equations 2 and 4 in Equation 1 and solving for L_{max}/D_0:

$$\frac{L_{max}}{D_0} = \sqrt{K_T(1 + p)\beta_2}\sqrt{\frac{\overline{U_0^2}}{gD_0\dfrac{\overline{T}_0 - T_a}{T_a}}} = \sqrt{K_T(1 + p)\,\beta_2}\,\sqrt{B_0} \quad . \quad (6)$$

TABLE 2—OUTLET FLOW DATA FOR 4-IN. *ASME* NOZZLE WITH WIDE OPEN DAMPER. DIAM. OF CORE = 1¾ IN.

$\tau_{0,\text{ no core}} = 0.40$ *lb* $\qquad Q_{0,\text{ no core}} = 4.0$ *cfs*

CORE SHAPE	$\frac{Q_0}{Q_{0,\text{ no core}}}$	$\frac{\tau_0}{\tau_{0,\text{ no core}}}$	β_2	C'_p	$\Delta\beta'_2$
No Core					
Core Bracket Attached Disc.	1.00	1.00	0.97	0	0
Core Hemispherical Core[a]	0.92	1.10	1.24	0.056	−0.069
Conical Core[a] Flat Base	0.96	1.10	1.16	0.001	−0.001
Conical Core[a] Hemispherical Base	0.96	1.08	1.14	−0.006	0.007
	0.96	1.10	1.16	−0.009	0.011

[a] Pointed up stream

$\beta_2 = \tau_0/[(Q_0/A_0)^2\,(\rho_0/g_c)\,A_0]$; $C'_p = (\overline{P'}_0 - P_a)/[Q_0/(A_0 - A_c)]^2\,(\rho_0/g_c)$

$\Delta\beta'_2$ = drag at free area of outlet/$[(Q_0/A_0)^2\,(\rho_0/g_c)A_0]$

Equation 5 states that when T_a, D_0, and ρ are fixed, $L^2_{max}\ (\overline{T}_0 - T_a)/K_T$ varies directly with the thrust. Equation 6 states that when D_0 and ρ are fixed, L^2_{max}/K_T varies directly with the product, $\beta_2\overline{B}_0$.

K_T is a dimensionless factor that decreases when the thermal spread of the jet increases radially; that is when the rate of temperature reduction in a radial direction out from the axis decreases. It follows that an increase in the thrust or an increase in $\beta_2\overline{B}_0$ will, in the absence of offsetting boundary effects□, produce either or both of two effects: (1) an increase in the throw; (2) an increase in the thermal spread of the jet either as the consequence of a decrease in K_T or as a consequence of an increase in $(\overline{T}_0 - T_a)/K_T$.

Equation 6 leads to the following expression for f:

$$f = [K_T\ (1 + \rho)\ \beta_2]^{1/2}/1.7 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

MEASURED VALUES OF β_2, THE OUTLET FLOW RATE, Q_0 AND THE OUTLET THRUST, τ_0

The following equations were employed to correlate the measured values of β_2. For the 4-in. long-radius *ASME* nozzle placed at the end of an 8-in. diameter duct:

$$\beta_2 = [\beta_{2,\ \text{no core}}][D^2_0/(D_0^2 - D_c^2)] \text{ for } D_c/D_0 < 0.8 \text{ and volume rates of flow within range covered in Fig. 7.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

where

$\beta_{2,\ \text{no core}} = 1.03$
$D_0 =$ outlet diameter.
$D_c =$ core diameter.

For the shallow diffuser attached to the unit heater and operated with the guide vanes removed and with disc cores:

$$\beta_2 = [\beta_{2,\ \text{no core}}][D^2_0/(D^2_0 - D^2_c)][1 - 0.37\ (D_c/D_0)^2] \text{ for } D_c/D_0 < 0.8 \quad . \quad . \quad (9)$$

where

$\beta_{2,\ \text{no core}} = 0.8.$

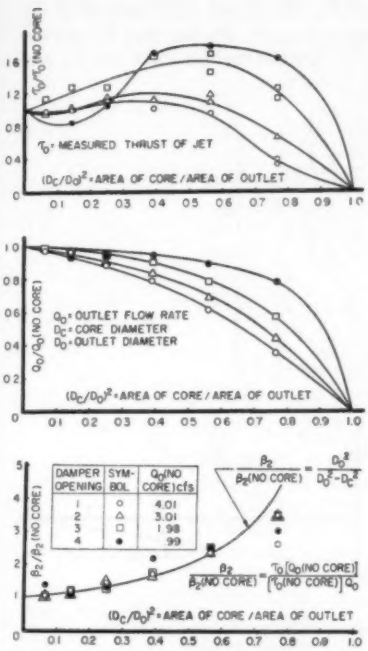
For the shallow diffuser attached to the unit heater and operated with guide vanes turned to different angles, α, from their perpendicular position with respect to the face of the outlet, and without cores:

$\beta_2 = 0.97 + 0.012\ \alpha$, for α between 0 and 60 deg.

Plotted values of β_2, Q_0 and τ_0 for the foregoing cases are presented in Figs. 7, 8 and 9. The curves of τ_0 and Q_0 are specific only for the unit heater employed and, in the case of the *ASME* nozzle, only for the fan and damper openings used. Since the fans were operated at constant speed, τ_0 had a maximum value. This occurred, not where β_2 was largest, but where $\beta_2 Q_0^2$ was a maximum.

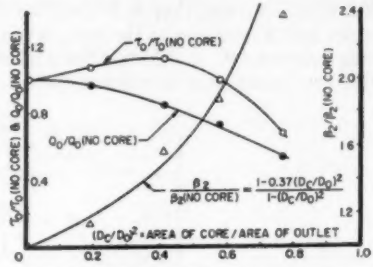
Table 1 gives data on β_2, Q_0 and τ_0 for both the straight cylindrical outlet and the shallow diffuser, each attached to the unit heater and equipped with (1) a 6-in. hemispherical core with the rounded face upstream, (2) a 6-in. disc core, and (3) no core. The values of β_2, in Table 1, average 0.78 for the cases wherein no core was employed and 0.88 for the cases wherein cores were employed.

Table 2 gives values of β_2, Q_0 and τ_0 for the 4-in. *ASME* nozzle equipped with cores having the various shapes shown in the table, and a diameter of 1¾ in. For the disc core, β_2 had a value of 1.24. For the hemispherical core, rounded face upstream, and the conical core pointed upstream, β_2 had an average value of 1.15.


T_0/T_0(NO CORE)
T_0 = MEASURED THRUST OF JET
$(D_C/D_0)^2$ = AREA OF CORE / AREA OF OUTLET
Q_0/Q_0(NO CORE)
Q_0 = OUTLET FLOW RATE
D_C = CORE DIAMETER
D_0 = OUTLET DIAMETER
DAMPER OPENING | SYMBOL | Q_0(NO CORE) cfs
1 | ○ | 4.01
2 | △ | 3.01
3 | □ | 1.98
4 | ● | .99
β_2/β_2(NO CORE)
$\frac{\beta_2}{\beta_2(\text{NO CORE})} = \frac{D_0^2}{D_0^2 - D_C^2}$
$\frac{\beta_2}{\beta_2(\text{NO CORE})} = \frac{T_0}{T_0(\text{NO CORE})} \left[\frac{Q_0(\text{NO CORE})}{Q_0}\right]^2$

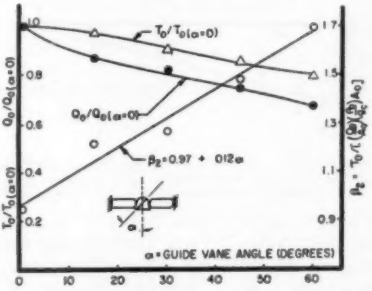
Values of relative measured outlet thrust, τ_0, outlet flow rate, Q_0, and relative thrust β_2, expressed as ratios of corresponding values obtained when no core was employed and plotted *vs.* ratio of core diameter squared, D_C^2, to outlet diameter squared, D_0^2. Note that maximum measured thrust occurs at different points for different damper settings and falls between $(D_C/D_0)^2$ equal to 0.3 and $(D_C/D_0)^2$ equal to 0.5. D_0 = diameter of core; D_0 = diameter of outlet. τ_0, no core for damper openings 1 to 4, respectively: 0.40 lb, 0.22 lb, 0.09 lb, and 0.025 lb. The curves of τ_0 and Q_0 are specific only for the outlet, fan and damper settings employed. The fan was operated at constant speed.

FIG. 7—OUTLET DATA FOR 4-IN. *ASME* NOZZLE WITH DISC CORES



Values of relative outlet thrust, β_2, outlet flow rate, Q_0, and measured thrust of jet, τ_0, expressed as ratios of corresponding values obtained when no core was employed are plotted *vs.* ratio of core area to outlet area, $(D_C/D_0)^2$
β_2/β_2, no core = $[\tau_0/\tau_0$, no core$]$ $[Q_0$, no core$/Q_0]$
Note that maximum value of τ_0 occurs when $(D_C/D_0)^2 = 0.4$. The curves are specific only for the unit heater and outlet employed. The fan was operated at constant speed.

FIG. 8—OUTLET DATA FOR SHALLOW DIFFUSER WITHOUT VANES AND WITH DISC CORES



(1) Values of outlet flow rate, Q_0, and measured thrust of jet, τ_0, expressed as ratios of corresponding values obtained when $\alpha = 0$, and (2) values of β_2 *vs.* guide-vane angle, α.
Q_0 ($\alpha = 0$) = 22.8 cfs (1370 cfm). τ_0 ($\alpha = 0$) = 0.70 lb.

FIG. 9—OUTLET DATA FOR SHALLOW DIFFUSER WITH GUIDE VANES AND WITHOUT CORES

The value of 0.97 for the *ASME* nozzle without core is low by about 6 percent, due to the core-holding device attached to the outlet.

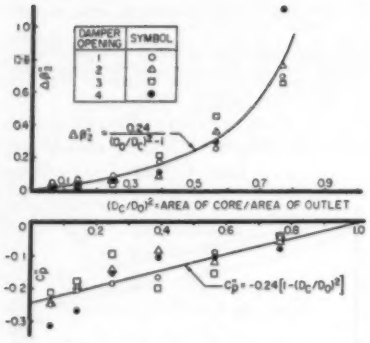
DRAG DUE TO NEGATIVE STATIC PRESSURES AT OUTLET

Let $\bar{P}'_0$ be the average absolute pressure at the annulus of a cored outlet and $\bar{P}''_0$ the average absolute pressure on the downstream side of the core. Then the drag, or negative contribution of $\bar{P}'_0$ to the projectile force of the jet, will be (P_a –

$\overline{P}'_0$) $(A_0 - A_c)$ and that of $\overline{P}''_0$, will be $(P_a - \overline{P}''_0)\,A_c$, where A_0 is the area of the outlet and A_c the area of the core. When there is no core, $\overline{P}'_0$ becomes $\overline{P}_0$ and the drag becomes $(P_a - \overline{P}_0)\,A_0$. For simplicity in correlating the data on $\overline{P}'_0$ and $\overline{P}''_0$, two terms were introduced:

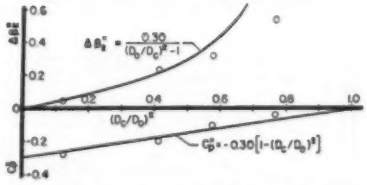
$$C_p' = \frac{-(P_a - \overline{P}'_0)}{\frac{\rho_0}{g_c}\left(\frac{Q_0}{A_0 - A_c}\right)^2} \quad \ldots \quad (10)$$

$$C_p'' = \frac{-(P_a - \overline{P}''_0)}{\frac{\rho_0}{g_c}\left(\frac{Q_0}{A_0 - A_c}\right)^2} \quad \ldots \quad (11)$$



$C''_p = (\overline{P}''_0 - P_a)/[(\rho_a/g_c)(Q_0/(A_0 - A_c))^2]$
$\Delta\beta'' =$ [drag at downstream face of core, lb]/[$((Q_0/A_0)^2(\rho_a/g_c)A_0$]

FIG. 10—*ASME* 4-IN. NOZZLE: VALUES OF $\Delta\beta_2''$ AND C_p'' FOR DISC CORES VS. RATIO, D_c^2/D_0^2, OF CORE DIAMETER SQUARED TO OUTLET DIAMETER SQUARED



$C''_p = (\overline{P}''_0 - P_a)/[(\rho_a/g_c)(Q_0/(A_0 - A_c))^2]$
$\Delta\beta''_2 =$ [drag at downstream face of core, lb]/[$((Q_0/A_0)^2(\rho_a/g_c)A_0$]

The upper solid curve was derived from the bottom solid curve using the relation:

$$\Delta\beta''_2 = -C''_p\,A_c A_0/(A_0 - A_c)^2$$

FIG. 11—SHALLOW DIFFUSER WITHOUT GUIDE VANES: VALUES OF $\Delta\beta''_2$ AND C_p'' VS RATIO D_c^2/D_0^2, OF CORE DIAMETER SQUARED TO OUTLET DIAMETER SQUARED

If the momentum force of the jet divided by $(\rho_0/g_c)(Q_0/A_0)^2(A_0)$ is denoted as β_1, then:

$$\beta_2 = \beta_1 - \Delta\beta'_2 - \Delta\beta''_2 \quad \ldots \quad (12)$$

where

$\Delta\beta'_2$ and $\Delta\beta''_2$ = the annulus and core-face drag components of β_2.

$\Delta\beta'_2 = -C'_p A_0/(A_0 - A_c)$.

$\Delta\beta''_2 = -C''_p A_c A_0/(A_0 - A_c)^2$.

Values of C'_p and $\Delta\beta'_2$ for the 4-in. *ASME* nozzle with 1¾-in. diam. cores were negligibly small. They are given in Table 2, based on $\overline{P}'_0$ extrapolated from the readings of 3 nozzle-wall pressure taps spaced 1 in., ½ in., and 1/16 in. from the face of the nozzle. Fig. 10 gives values of C''_p and $\Delta\beta''_2$ for disc cores placed in

the outlet of the *ASME* nozzle. The curves plotted in Fig. 10 may be represented by:

$$C''_p = -0.24\,[1 - (D_e/D_0)^2] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

$$\Delta\beta''_2 = 0.24/[(D_0/D_e)^2 - 1] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

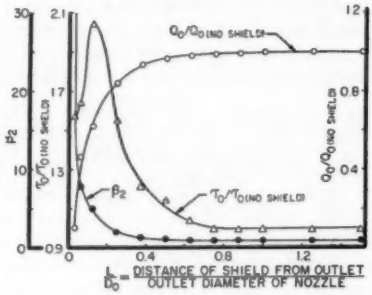


FIG. 12—*ASME* 4-IN. NOZZLE: VALUES OF THE OUTLET RELATIVE THRUST, β_2, THE MEASURED OUTLET THRUST, τ_0, AND THE OUTLET FLOW, Q_0, VS. THE DISTANCE OF SHIELD, L/D_0, FROM THE OUTLET WHEN A 33-IN. SQUARE SHIELD WAS PLACED VERTICALLY IN THE JET. SHIELD HAD NO EFFECT ON OUTLET CONDITIONS WHEN L/D_0 EXCEEDED 1

Fig. 11 gives values of C''_p and $\Delta\beta''_2$ for disc cores placed in the outlet of the unit heater equipped with the shallow diffuser with guide vanes removed. The curves plotted may be represented by the equations:

$$C''_p = -0.30\,[1 - (D_e/D_0)^2] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

$$\Delta\beta''_2 = 0.30/[(D_0/D_e)^2 - 1] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

For evaluating C''_p and $\Delta\beta''_2$, $\bar{P}''_0$ on the downstream face of the disc cores was evaluated as the arithmetical mean of the average readings of four 0.056-in. I.D. pressure taps located at the corners of a 2-in. square centered on the disc.

Table 1 gives values of C''_p and $\Delta\beta_2''$ for the cylindrical outlet and shallow diffuser as employed, each equipped with (1) a 6-in. hemispherical core with the rounded face upstream and (2) a disc core. $\bar{P}''_0$ was evaluated as the arithmetical average of three 0.056-in. I.D. pressure taps located on the downstream face of the core 7/8 in., 1 11/16 in., and 2 23/32 in. from the center of the core.

Because of the uncertain accuracy of static pressure readings made in the presence of whirl, no attempt was made to measure $\bar{P}'_0$, the average static pressure in the annulus, when dealing with nozzles attached to the unit heater.

THE QUANTITY $[(\Sigma U^2)/N]/[(\Sigma U)/N]^2$ AND RELATIVE MOMENTUM FLUX, β_1

The quantity $[(\Sigma U^2)/N]/[(\Sigma U)/N]^2$ is an index of the degree of nonuniformity of velocity at the face of the outlet. The numerator in this quantity is the average of

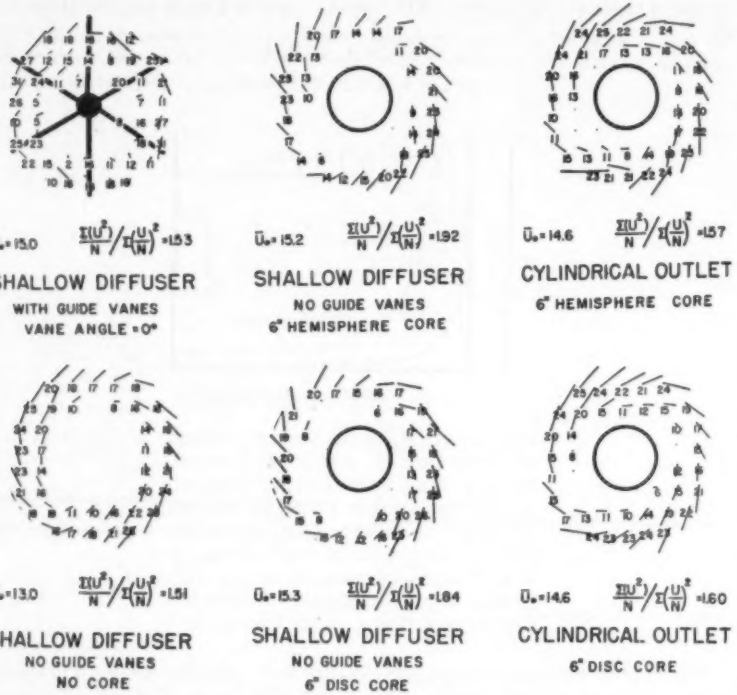


FIG. 13—PHOTOGRAPHS OF 3-DIMENSIONAL MODELS OF OUTLET VELOCITY VECTORS LOOKING INTO OUTLETS. THE NUMBER ASSOCIATED WITH EACH VECTOR REPRESENTS THE MAGNITUDE OF THE VECTOR. PRACTICALLY ALL VECTORS POINT IN THE DIRECTION OF CLOCKWISE ROTATION, INDICATING CLOCKWISE WHIRL. N = NUMBER OF READINGS TAKEN; U = LOCAL VELOCITY AT OUTLET; $\bar{U}_0$ = AVERAGE VELOCITY AT OUTLET

N (56, two in. apart) separately squared velocity readings at the outlet. The denominator is the square of the arithmetical average of the N readings. The quantity $[(\Sigma U^2)/N]/[(\Sigma U)/N]^2$ may be regarded as a rough approximation of β_1. Values of $[(\Sigma U^2)/N]/[(\Sigma U)/N]^2$ for the outlets attached to the unit heater appear in Table 1 and Fig. 13.

β_1 for the *ASME* nozzle without core was 1.03. For the cored *ASME* outlets, β_1 was evaluated as $\beta_2 + \Delta\beta''_2$, $\Delta\beta'_2$ being negligible. This gave:

$$\beta_1 = [D_0^3/(D_0^3 - D_c^3)][1.03 + 0.24\ (D_c/D_0)^2] \quad . \quad . \quad . \quad . \quad . \quad (17)$$

EFFECT OF JET'S IMPINGEMENT ON VERTICAL PLATE

Fig. 12 shows the effect on the outlet relative thrust, β_2, the outlet thrust, τ_0, and the outlet flow, Q_0, produced by a 33-in. square plate or shield placed vertically

in the jet from the *ASME* nozzle at various distances, L/D_0, from the outlet. τ_0 and Q_0 are plotted as fractions of their respective values before the insertion of the shield. When the shield was more than one outlet diameter from the outlet, it had no significant effect upon outlet conditions. The flow rate before the shield was inserted into the jet was 4 cfs (240 cfm).

A study was made of the effect on the thrust produced by: (1) a 2-ft x 3-ft cardboard flange attached to and supported by the outlet flush with the outlet face, and (2) the same flange supported in the same position but free from contact with the nozzle. The effect of this flange was in both cases negligible.

VELOCITY DISTRIBUTION AT THE OUTLET

Fig. 13 is a pictorial representation of the pattern of velocity distribution at the outlet of the unit heater for each of 6 cases. In 4 of these, different modifications of the shallow diffuser were employed as the outlet. In the 2 remaining cases different modifications of the cylindrical outlet were employed. In preparing Fig. 13, a model was first made of the velocity distribution in each of the 6 cases, and then photographed. The model was 8¾ in. in diameter. Photographs were taken from a distance approximately 8 ft from the model. Velocities were represented in the model, both in magnitude and direction, by segments of 1/16-in. welding rod. The number in the figure associated with each segment gives the magnitude of the velocity to the nearest whole number. Velocities at the outlet were measured at 56 points spaced 2 in. apart. Velocities in the vicinity of the center of the outlet were in all cases zero.

EFFECT OF CORES ON DOWNSTREAM VELOCITY DISTRIBUTION

Fig. 14 shows the velocity distributions in the jet from the *ASME* nozzle at various distances from the outlet with and without a 2-in. disc core placed at the nozzle outlet. The air flow for the case in which the core was present was approximately 13 percent lower than the flow without the core. Within the accuracy of the instrumentation, the core had no effect on the total jet angle (approximately 18 deg), nor on the shape of the velocity profile after the flow had reached a distance of about 3 outlet diameters from the outlet of the nozzle.

Fig. 15 shows the velocity profiles that were obtained with each of 4 different 1¾-in. cores placed in the *ASME* nozzle outlet. At 3 outlet diameters from the nozzle outlet, and beyond, the pattern of the velocity distribution was essentially the same for the 4 cases. The jet angle for the 4 cores was between 18 and 19 deg.

EFFECT OF WHIRL

The following ratios will be employed:

$$(\beta_1/\beta_2) = \text{(Momentum Force at Outlet)/(Measured Thrust)} \quad . \quad . \quad . \quad . \quad . \quad (18)$$

$$(\Delta\beta'_2/\beta_2) = \text{(Negative-Pressure Drag at Free Area of Outlet)/(Measured Thrust)} \quad (19)$$

$$(\Delta\beta''_2/\beta_2) = \text{(Negative-Pressure Drag at Downstream Face of Core)/(Measured Thrust)} \quad . \quad . \quad . \quad . \quad . \quad (20)$$

In the absence of deflecting devices, significant negative pressures will be present at the free area of an outlet only when produced by whirl, eddies, or other similar

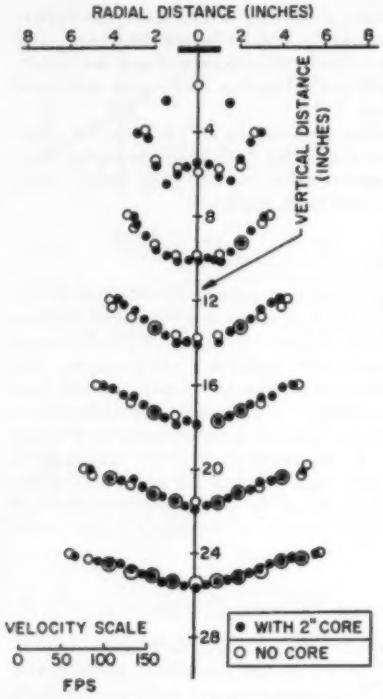


FIG. 14—DOWNSTREAM VELOCITY DISTRIBUTION IN JET FROM 4-IN. *ASME* NOZZLE, WITH AND WITHOUT A 2-IN. DISC CORE. VELOCITY AT AXIS WAS SLIGHTLY GREATER WHEN THE CORE WAS EMPLOYED, BUT PATTERN OF DISTRIBUTION DOWNSTREAM FROM OUTLET BECAME SIMILAR FOR THE 2 CASES

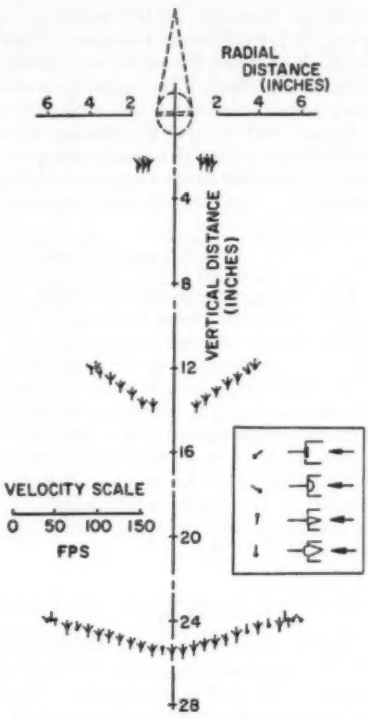


FIG. 15—DOWNSTREAM VELOCITY DISTRIBUTIONS IN JET FROM THE *ASME* 4-IN. NOZZLE WITH 4 DIFFERENT TYPES OF 1¾-IN. CORES PLACED IN OUTLET IN THE DIRECTION INDICATED BY DIAGRAMS

effects. For outlets without deflecting vanes, $\Delta\beta'_2/\beta_2$ may be regarded, therefore, as the relative drag due to whirl.

The value of $\Delta\beta'_2/\beta_2$ for the *ASME* nozzle is zero. Values of $\Delta\beta'_2/\beta_2$ for the cylindrical outlet and the shallow diffuser evaluated from the data in Table 1 on the assumption that $\beta_1 = (\Sigma U^2/N)/(\Sigma U/N)^2$ are given in Table 3 together with corresponding values of β_1/β_2 and $\Delta\beta''_2/\beta_2$.

Values in Table 3 indicate that the drag effects due to whirl were equal to 80 to 100 percent of the net thrust or 40 to 50 percent of the momentum force of the jet.

The effect of whirl is furthermore demonstrated by the fact that both the core-free shallow diffuser and the core-free cylindrical outlet when attached to the unit

heater and operated without guide vanes or any other device for checking whirl, developed values of β_2 substantially less than unity, namely 0.77 and 0.79. When vanes set perpendicular to the plane of the outlet were employed in the shallow diffuser, the value of β_2 increased from 0.79 to 0.97.

RECAPITULATION

1. A simple means for measuring the thrust developed by an outlet and evaluating one component, $\beta_2^{1/2}$, of the throw factor, f, in the equation $L_{max}/D_0 = 1.7 f \bar{B}_0^{1/2}$ have been presented together with typical data on these quantities.

TABLE 3—TABULATION OF BETA RATIOS

	(1)	(2)	(3)	(4)
	$\left(\frac{\text{MOMENTUM FLUX}}{\text{THRUST}}\right)$	$\left(\frac{\text{WHIRL DRAG}}{\text{THRUST}}\right)$	$\left(\frac{\text{CORE DRAG}}{\text{THRUST}}\right)$	$\left(\frac{\text{COL. 2}}{\text{COL. 1}}\right)$
	β_1/β_2	$\Delta\beta'_1/\beta_2$	$\Delta\beta''_2/\beta_2$	
Cylindrical Outlet 6 in. core no vanes	1.89	0.83	0.06	0.44
Shallow Diffuser 6 in. core no vanes	2.10	1.05	0.05	0.50
Shallow diffuser no core no vanes	1.92	0.92	0	0.48

2. Modifications to an outlet of fixed diameter which increase the product, $\beta_2 \bar{B}_0$, have been shown to result in either a longer downthrow or a wider radial distribution of temperature effects, or in both effects each to a lesser extent, provided the increase in $\beta_2\bar{B}_0$ is not offset by boundary effects.

3. Whirl at the face of typical outlets created a substantial negative-pressure drag which resulted in values of thrust and β_2 well below the values that would have been realized in the absence of whirl. Guide vanes reduced whirl and thereby the drag due to negative pressures at the face of the outlet.

It is suggested that the means described herein for evaluating outlet characteristics be given consideration by manufacturers of unit heaters. These means may be employed in local laboratories where space with high ceilings, such as is needed for full-scale tests, may not be available.

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NOMENCLATURE

A = area, square feet.
A_c = area of core, square feet = $\pi D_c^2/4$.
A_0 = area of outlet, square feet.
A_s = cross-sectional area of heated space, square feet.
B_0 = cross-stream average buoyancy number at outlet = $(\overline{U}_0^2/gD_0)/[(\overline{T}_0/T_a) - 1]$.
C'_p = $(\overline{P}'_0 - P_a)/(\rho_0/g_c)[Q_0/(A_0 - A_c)]^2$.
C''_p = $(\overline{P}''_0 - P_a)/(\rho_0/g_c)[Q_0/(A_0 - A_c)]^2$.
D_c = diameter of core, feet.
D_0 = diameter of outlet, feet.
f = throw factor in equation for L_{max}/D_0.
ΔF_b = resultant of vertical forces generated by jet at the boundaries of the space heated (positive downward), pound.
g = 32.2 feet per (second) (second).
g_c = 32.2 (pound-mass) (feet) per (pound-force) (second) (second).
K_T = spacial temperature distribution factor.
K'_T = shape factor for temperature profile in equation:

$$(T - T_a)/(T_c - T_a) = e^{(-K'_T r^2/L^2)}$$

L = distance from outlet along center line of nozzle projected, feet.
L_{max} = distance from outlet to bottom of jet.
L_p = an empirical constant from the equation: $(\overline{T}_0 - T_a)/(\overline{T}_0 - T_a) = L_p/L$
N = number of velocity measuring stations at the outlet of the unit heater.
$_0$ = subscript referring to outlet.
p = ΔF_b/[outlet downthrust]
P_a = atmospheric pressure, pounds per square foot, absolute.
$\overline{P}_0$ = average static pressure at outlet, pounds per square foot, absolute.
$\overline{P}'_0$ = average static pressure of the air stream at the outlet, pounds per square foot, absolute.
$\overline{P}''_0$ = average static pressure on downstream face of core, pounds per square foot, absolute.
Q_0 = air flow at outlet, cubic feet per second.
r = radial distance from axis of jet, feet.
T_a = reference room temperature, Fahrenheit degrees, absolute.
T_{av} = mass-mean temperature of air in heated space, Fahrenheit degrees, absolute.
T_c = Temperature on vertical axis of jet, Fahrenheit degrees, absolute.
$\overline{T}_0$ = mass-mean temperature at outlet, Fahrenheit degrees, absolute.
U = local velocity, feet per second.
$\overline{U}_0$ = cross-stream average velocity at the outlet, feet per second = Q_0/A_0.
a = guide-vane angle = angle between plane of blade and plane perpendicular to face of outlet.
β_1 = [outlet momentum force (flux)]/

$$\left(\frac{Q_0}{A_0}\right)^2\left(\frac{\rho_0}{g_c}\right)A_0 = \frac{\int_{A_0} \frac{\rho_0 U_0^2}{g_c}\,dA}{\left(\frac{Q_0}{A_0}\right)^2\left(\frac{\rho_0}{g_c}\right)A_0}.$$

β_2 = (measured thrust of jet)/$(Q_0/A_0)^2(\rho_0/g_c)A_0$ or $[\rho_0\overline{U}_0^2 + g_c(P_0 - P_a)]/[\rho_0\overline{U}_0^2]$.
$\Delta\beta'_2$ = (relative drag at free area of outlet)β_2.
= (negative-pressure drag at free area of outlet/measured thrust) β_2.
$\Delta\beta''_2$ = (relative drag at downstream face of core)β_2.
= (negative-pressure drag at downstream face of core/measured thrust)β_2.
ρ_a = density of air corresponding to T_a, pounds per cubic foot.
ρ_0 = mass density at outlet, pounds per cubic foot.
ρ_{av} = average mass density of air in heated space, pounds per cubic foot.
τ_0 = directly measured thrust, pounds force.

APPENDIX

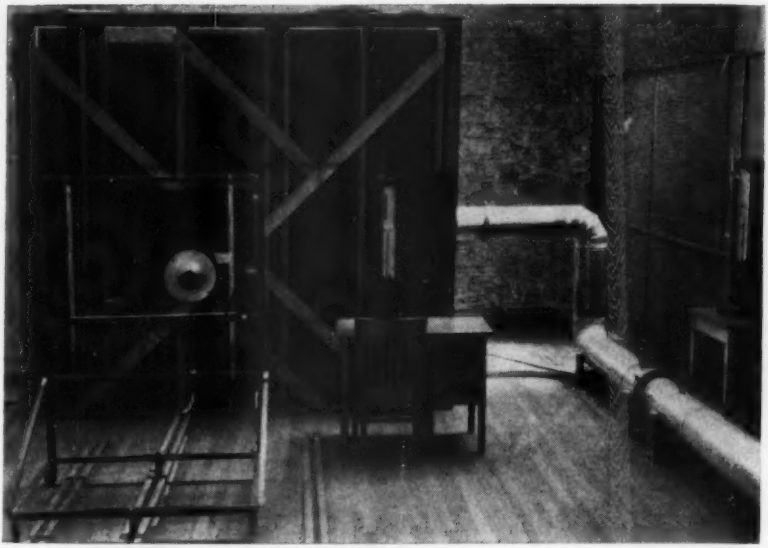

FIG. A-1—TEST STATION SHOWING INSTRUMENT CARRIAGE IN FOREGROUND AND BACK OF IT THE *ASME* NOZZLE MOUNTED ON THE SWING PLATFORM (NOT VISIBLE). THE DUCT AT THE RIGHT LEADS FROM THE EXTERNAL FAN TO THE HOUSING FOR THE SWING PLATFORM

DISCUSSION

L. R. PHILLIPS, Hartford, Conn. (WRITTEN): This paper is a continuation of the excellent work on heated jets conducted by Professor Helander and his associates. It encompasses a wealth of information on 3 different aspects of the subject. First, the characteristics of the jet from experimental observations and the formulation of the information into an orderly empirical, usable form; second, the theoretical and mathematical coordination of these data with the analytical approach as it now stands; and third, a detailed description of a good experimental method and novel instrumentation.

The paper is so full of this 3-fold, solid information, that, contrary to so many papers, it lets itself down in the fact that it does not have any applicable content of self-advertising and promotional material. For example, how valuable are these data, who can use them, in what manner and with what beneficial results?

As one vitally interested in the problem from the aspect of air diffuser or outlet design, I feel that the use of the thrust parameter as developed in this paper will be of significant importance. It is useful, practical information, from which a formula is derived, making it applicable to equipment differing from the specific units used for the testing.

The experimental method developed for these tests is also a practical tool, whose use, I believe, may be extended to test other types of outlets, for example those ordinarily used for projection heating.

APPENDIX *(continued)*

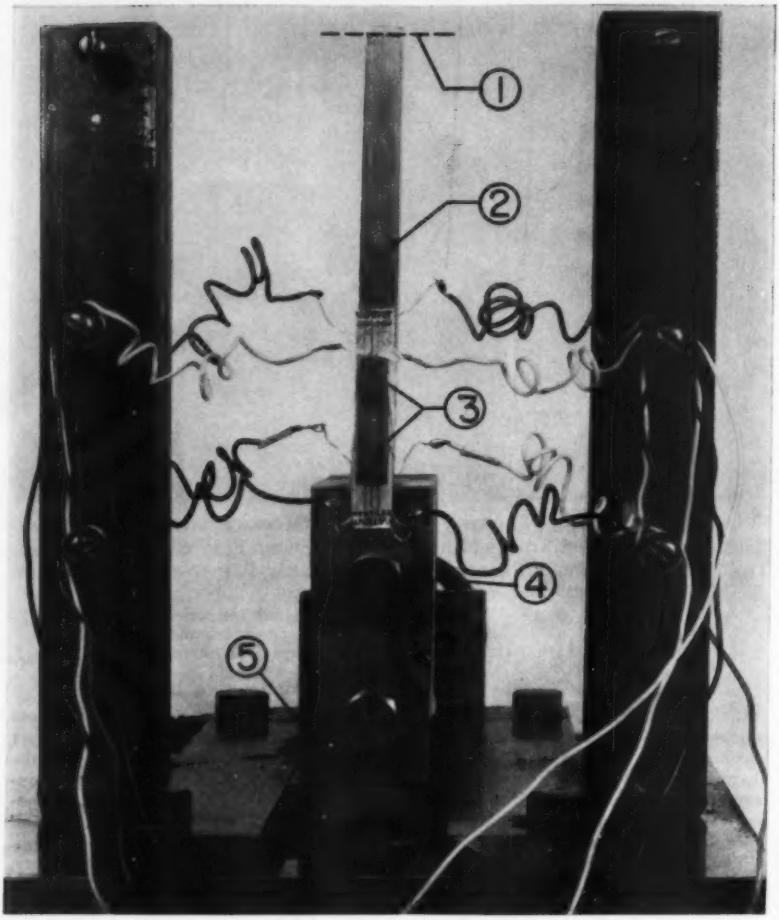

FIG. A-2—HEAD-ON VIEW OF THE THRUST MEASURING DEVICE

Reference Numbers: (1) line of contact between swing platform and cantilever beam, (2) 6 x ½ x ⅛-in. aluminum cantilever beam, (3) SR-4 type A5 strain gages, (4) adjusting wheel, for pressing cantilever-beam (2) against swing platform so as to restore plumb position. (5) base adjustable in line parallel to direction of thrust. An SR-4 type M strain indicator was employed.

I hope that Professor Helander and his colleagues Professors Yen and Tripp continue this investigation and that in the near future, find time to prepare a paper summarizing and coordinating all the work on this subject done at Kansas State, in a form such as might be adaptable for inclusion in THE GUIDE.

H. B. NOTTAGE, Encino, Calif. (WRITTEN): May we have a complete statement from the authors covering the basic problems yet to be solved in connection with their non-isothermal jet system, including proposed procedures for performing the needed research? What is the continuing plan at Kansas State College?

W. O. HUEBNER, New York, N. Y.: It is always a privilege and a pleasure for the chairman of a Technical Advisory Committee to listen to the presentation of a paper which has been prepared under the guidance of the T.A.C.

There may be people in the audience to whom papers of this type seem to be too much on the theoretical side but, it is my earnest conviction, that the Kansas State Research Project on Downthrow of Heated Air Jets is of the greatest practical value. The helpful nomogram published in the January 1957 issue of *Heating, Piping & Air Conditioning* (page 187), which is based upon the findings of Professor Helander and his associates, proves this point. The nomogram helps to find the outlet area for the downward jet of heated air if the jet length, the temperature differential and the cfm are given, and has been prepared by Messrs. W. W. Kennedy and E. C. Varum. Mr. Kennedy is an active member of the T.A.C. on Air Distribution.

I am particularly happy that Professor Helander has told us that he will prepare a paper summarizing the results of his work. I believe this is pioneering work of which the Society can be extremely proud.

J. J. AEBERLY, Chicago, Ill.: I did not intend to discuss this paper. However, two names were mentioned here, Kennedy and Varnum, which recall to my mind an investigation and study made in Chicago regarding short circuiting of air when a single fixture is used for supply and return. The needs of the Chicago study clearly show the importance of accurate formulae and methods to determine values, as these authors have presented them in this paper. We wanted to be sure that the minimum quantity of air required by the Municipal Code reached the breathing zone fixed below the 6-ft level of the room. To accomplish this, the designer was required to increase the deliveries 100 percent and was also told that this value could be lowered if accurate laboratory techniques were followed and substantiating data for a lower value were submitted and approved. Kennedy was intimately connected with the Chicago study. The rules of thumb and the knowledge gained from experience were found to be insufficient. Exact scientific analysis and difficult mathematical calculations were needed, just as is evident in the authors' paper. The empirical values obtained in our study were excellent. We were surprised, however, to find that our consideration, because of the needs of our problem, compels us to use such basic values in chemistry and physics as the conservation of energy and the conservation of momentum.

It should be remembered, that the speed of a single molecule of hydrogen on a free path and at ordinary room temperature, is approximately 40,000 fpm. This speed changes as it meets with interference of other molecules, the mass movement of other gases and dust, temperature changes and rarification of gases at exhaust outlets of the room. Values obtained by approximations are unreliable. These problems require that calculations be made using logarithmic progression and at times difficult procedures in calculus.

Although we should strive for simplification in our analytical presentation, we must face it, there are no rules of thumb or short cut approaches. The authors' methods are sound, even though we are set to wondering. When the empirical values, which were relevant were analyzed, and the accepted values published in THE GUIDE were used, some of us reached the conclusion that short circuiting was zero. I could not agree with these findings. Yesterday, Hemeon presented a paper in which he referred to the

velocity contours determined by Dala Valle, shown on page 1043 of THE GUIDE, values which served as the initial basis for my rejection. However, before I could prove my contention and establish the value of 25 percent to 30 percent short circuiting, I was forced to deal with this problem in terms of instantaneous values, and make use of some difficult procedures in calculus. You should not be disturbed because of short circuiting in the use of this type of fixture. Here we have the problem of a point of diminishing return. The reduction of cost of partition changes in large offices and commercial buildings, frequently more than compensates for the loss in efficiency in delivering the air supply. I will be interested in studying further the methods and values presented in this paper and others of this kind.

AUTHORS' CLOSURE (Mr. Helander): We are grateful to Mr. Aeberly, Mr. Huebner, Dr. Nottage, and Mr. Phillips for the contributions they have made in the form of discussions of the paper. Mr. Aeberly, in describing his own experiences, has stressed what needs to be emphasized, namely that (1) mathematics and the principles of fluid mechanics are essential tools for interpreting and correlating data on air distribution; and (2) there is a limit to the order of simplification that may be employed in interpreting and correlating such data.

Mr. Phillips, Dr. Nottage, and Mr. Huebner have expressed an interest in the direction our future research will take. We are now engaged in exploring the distribution of temperatures and velocities in the entire space affected by the jet. This space extends well beyond the boundaries of the downwardly moving jet. Our next paper will deal with the results obtained. In addition, this paper will summarize and interpret the results of all of our studies in a manner which we hope will be found useful. Our plan is to present the data in the form of empiricisms consistent with the principles of mechanics and, in so doing, employ the dimensionless criteria $\overline{B}_0$, β_2, and K_T. Our next paper will conclude our studies of jets projected downward from outlets of the type used in practice. However, our interest in the theory of jets and in fundamental research on jets will continue.

1602

No. 1602

AIR FLOW IN FREE CONVECTION OVER HEATED BODIES

By T. F. HATCH* AND D. BARRON-ORONZCO**, PITTSBURGH, PA.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, in cooperation with the Graduate School of Public Health, University of Pittsburgh, Pittsburgh, Pa.

THE RATE of local exhaust ventilation required for the capture and removal of atmospheric contaminants released by industrial processes is primarily determined by the magnitude of air flow away from the process rather than by characteristics of generation and release of the contaminant itself. This is because fine dust particles, fumes, vapors and gases given off by the process are not dispersed *through* air†. They mix intimately with the air immediately around the point of release and are carried away only by air movement. Thus, the stream of air flowing past the point of release picks up the contaminant and it is this stream of contaminated air which must be captured and removed. From this it follows that the starting point in the determination of the local exhaust ventilation requirement for a given process is an identification of the source or sources of air motion which are operating around the process, the determination of the patterns of air motion, such as direction and velocity of flow, and estimation of the overall rate of flow of the contaminated air away from the process.

In certain categories of industrial processes, the sources of the troublesome air motion are inherent in the process itself. The forces operating to create the air flow are well identified and can be related to measurable characteristics of the process. If such relationships can be developed in systematic physical and mathematical terms, the design engineer is thereby provided with important information to guide him in his determination of the local exhaust ventilation requirements.

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† Relatively coarse particles, released with high initial velocity, are projected through air but those of principle hygienic interest (<10 u) are not. Vapors and gases are also dispersed through air by molecular diffusion, but the diffusion velocity is relatively low compared with normal velocities of air movement even in "still air."

Presented at the 63rd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Chicago, February 1957.

An important group of industrial processes which inherently create air motion are those involving release of heat. The process equipment operates at an elevated temperature and strong vertical convection currents are created. Any contaminant given off by the process will be incorporated into the rising stream of heated air and be dispersed with it‡. Local exhaust ventilation, such as by the common canopy hood, must be applied at a rate in excess of the rate of convection flow to ensure successful control of the contaminant.

PHYSICAL RELATIONSHIPS

As a buoyant stream of heated air rises it acts like a jet and, by turbulent mixing, draws cooler surrounding air into the rising stream. In consequence, the volume rate of flow increases with distance above the heat source. Since the amount of heat in the stream remains constant, the excess temperature above the ambient level decreases as distance increases and the velocity decreases. The cross-sectional dimensions of the stream expand as the heated air rises. When the excess temperature drops below a certain value the buoyancy of the air is used up by turbulent mixing and the upward direction of flow is destroyed. The rising stream terminates by complete mixing with the ambient air.

Despite the practical importance of free convection air flow in many situations involving heat transfer, the relationships involved in this phenomenon have received very little attention by physicists and engineers. The problem was dealt with by Schmidt[1], who derived a mathematical expression from theoretical considerations and confirmed by some limited experimental data. Sutton[2] has considered the problem on a theoretical basis, particularly in respect to the stream of hot gases issuing from a chimney. This has great practical value in problems of community air pollution and is of interest here since it should have equal application to the free convection flow created by a hot body.

On the basis of mixing theory, Sutton derived the following expressions for the change in stream velocity and in radius with distance above a hypothetical *point source* of heat:

$$W = W_1 Z^{-M/6} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

$$W_1 = [7g/(3\pi C_p \rho T_a C)]^{1/3} h^{1/3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

$$r = 1.5 CZ^{M/2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where

W = stream velocity, feet per second.
W_1 = constant, for given rate of heat output, h.
Z = distance above *point source*, feet.
h = convective heat output from source (heat carried away by rising stream) (Btu/sec).
g = gravitational constant, 32.174 ft per (sec) (sec).
C_p = specific heat of air at constant pressure, Btu per (pound) (Fahrenheit degree).
ρ = density of air at ambient temperature and pressure, pounds per cubic foot.

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‡ The heat itself may be considered an atmospheric contaminant and local exhaust ventilation is frequently applied to hot process equipment in order to capture and remove the heated air before it escapes into the surrounding work space.

[1] Exponent numerals refer to References.

T_a = temperature of ambient air, Fahrenheit absolute.
C = coefficient of mixing by eddy diffusion.
r = radius of rising stream of heated air, feet.

From an analysis of Schmidt's experimental data, Sutton concluded that the exponent, M, has a value of 1.75 (Schmidt took $M = 2.0$) and from his own work on the dispersion of gases with distance from a point source, he found $C = 0.17$ $ft^{1/8}$. In English units, converting velocity to feet per minute, V, and convective heat flow to the common form, H, Btu/hr:

$$V_1 = [(7 \times 32)/(3\pi \times 0.25 \times 0.075 \times 530 \times 0.17)]^{1/3} (H/3600)^{1/3} = 0.16\, H^{1/3}$$

$$r = 1.5 \times 0.17\, Z^{0.88} = 0.255\, Z^{0.88}, \text{ ft} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

$$A_s = \pi r^2 = 0.203\, Z^{1.75}, \text{ sq ft}$$

$$V = 60 \times 0.16\, H^{1/3} Z^{-0.29} = 9.6\, H^{1/3} Z^{-0.29}, \text{ fpm} \quad . \quad . \quad . \quad . \quad . \quad (5)$$

and, since

$$Q = VA_s$$

$$Q, \text{ in cfm} = 1.95\, H^{1/3} Z^{1.46}$$

where

H = Btu per hour
Z = feet.

Here, it is noted that the rate of flow of the rising stream at distance Z above the hypothetical point source is proportional to a constant, to the 1/3 power of the convective heat output and to the 1.46 power of the distance. Since a point source of heat release was assumed, no physical dimensions of the heated body appear in the equation.

APPLICATION OF SUTTON'S EQUATION TO LOCAL EXHAUST DESIGN

In practical application of local exhaust ventilation over hot process equipment, the heat source has real dimensions and the contaminated air must be captured within a relatively short distance above the process in order to keep the required rate of ventilation at a minimum and to minimize opportunity for spread of the contaminant into the general atmosphere. A hot body of real size may be treated theoretically as a point source of heat beyond a certain minimum distance but Sutton's equation requires experimental verification to demonstrate its practical usefulness within the relatively short distances at which local exhaust hoods are applied above hot process equipment. This was the primary objective in the present study.

For purposes of this study, the equation has been rewritten to give a linear relationship, the distance Z being expressed as: $Z = A + X$ where A is the distance from the hypothetical point source of heat to the top surface of the hot body and X is the distance from the top surface to the level where local exhaust ventilation is to be applied (see Fig. 1). Then,

$$(Q/H^{1/3}) = K(A + X)^{1.46}$$

$$(Q/H^{1/3})^{1/1.46} = K^{1/1.46} (A + X) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

Equation 6 is in the form of a linear relationship and should give a straight line on an arithmetic grid when $(Q/H^{1/3})^{1/1.46}$ is plotted against X. The slope of this line will give the value of the constant, $K^{1/1.46}$ and the intercept at $X = 0$ should equal $AK^{1/1.46}$.

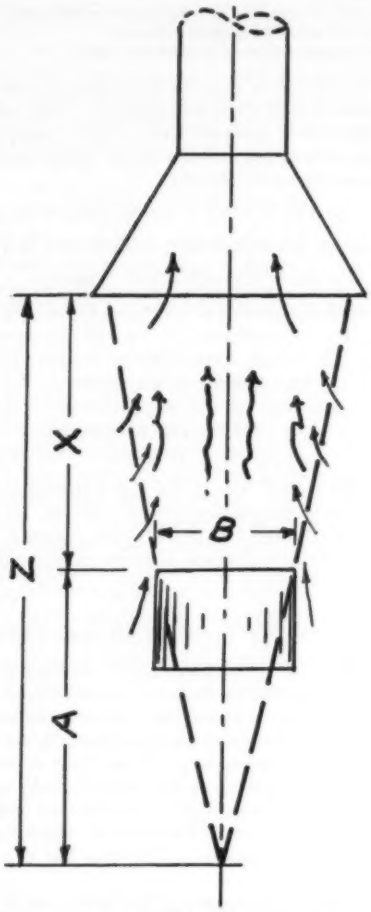
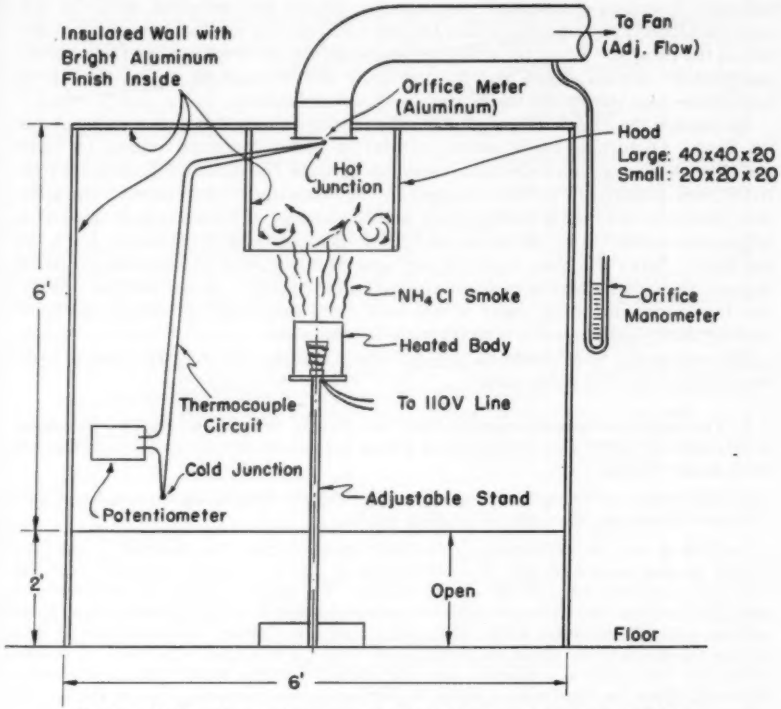


FIG. 1—PATH OF EXPANDING STREAM OF HEATED AIR, SHOWING RELATION TO THE THEORETICAL POINT SOURCE AND TO DISTANCE, Z

Thus, a ready means is at hand for testing the application of the equation within short distances, X, from experimentally determined values of Q *vs.* X for various values of heat output, H. One may also determine the influence of the dimensions of hot bodies of different sizes upon the value of A or other influence upon the relationship. It is to be noted in this connection, that the radius of the rising stream increases with the 0.88 power of distance. If, at the top surface of the hot body,



Aluminum lined enclosure and arrangement of heated body, receiving hood and air flow meter for determining convection air flow in relation to distance above heated body.

FIG. 2—EXPERIMENTAL SETUP FOR DETERMINING AIR FLOW BY FREE CONVECTION OVER A HEATED BODY

the rising stream is assumed to have the same cross-sectional dimension as the body B, then, from Equation 4:

$$A = 2.15\ B^{1.14}$$

EXPERIMENTATION

The physical facilities for determining the variation in Q with X are shown in Fig. 2. The experimental hot body, consisting of an electrical resistance heater inside a metal can, was mounted on an adjustable stand axially below an exhaust hood extending down from the ceiling of an insulated cubicle placed in the center of a 24-ft square laboratory. The exhaust hood was connected to a variable output fan, the air flow being measured by a sharp-edge orifice placed at the inlet to the exhaust pipe with the orifice head indicated by a water manometer in the standard

manner. A copper-constantan thermocouple circuit was provided with the hot junction placed just below the orifice and the cold junction located at a remote point within the cubicle. From the temperature rise in the air stream above the ambient temperature thus measured, and the rate of air flow through the orifice, the rate of convective heat output by the heated body was calculated.

Air entered the cubicle at a negligible velocity through the 2-ft high opening above the floor. Disturbing crosscurrents within the cubicle were minimized by tight walls which also provided thermal insulation to make the cubicle temperature relatively unresponsive to sudden changes in the laboratory temperature. In addition, the walls and ceiling of the cubicle and the interior of the hood were lined with bright aluminum foil to minimize heating of these surfaces by radiation from the hot body. Since the body operated with a surface temperature of several hundred degrees, the radiant heat output was at least twice the output by convection. Without the aluminum lining, there would have been significant development of disturbing down and cross-drafts in the experimental room.

Measurements were made in the following manner for a given heated body mounted on an adjustable stand:

1. The rate of air flow through the hood was first set to a desired level by the choice of a proper size orifice and adjustment of a blast gate in the duct to give the appropriate manometer reading.

2. The stream of rising hot air was made visible by volatilizing a quantity of ammonium chloride on the body to produce smoke.

3. With Q and the heat input to the body kept constant, the distance X was then varied up and down until the distance was found where no smoke was seen to escape outward under the edge of the exhaust hood. To test this end-point sensitively, a strong light beam was directed parallel to and just below one edge of the hood. There was no other illumination within the cubicle and the observer, standing quietly in a position opposite to the illuminated edge, could note the direction of travel of the smoke. When the end-point was reached, he could detect a rather stable volume of smoke-filled air filling the hood with a fairly well defined lower boundary just at the bottom edge of the hood. It was necessary for him to remain quiet since the smallest cross-current tended to upset the flow pattern and cause momentary escape of smoke. For each setting of X it was necessary to observe the performance over a sufficient period of time to allow the system to stabilize and the thermal gradient to become steady.

4. When the proper distance X had been determined, the temperature rise was measured. It was found that the addition of ammonium chloride smoke to the air stream increased the convective heat output and thus disturbed the Q *vs.* X relationship. In order to minimize this source of error, the amount of ammonium chloride added in each test was standardized and kept approximately constant. Furthermore, the thermocouple reading was not taken until after the chloride had completely volatilized and the air stream was clear.

5. For a given body with fixed heat input, the foregoing series of steps was repeated for different values of Q and the corresponding values of X determined over a range from a fraction of the hot body diameter up to 5–6 diam distance between the heated body and the hood.

6. Four different sizes of heated bodies were studied: (a) 5½-in. diam x 6 in. long cylinder, observed with axis both vertical and horizontal; (b) 5½-in. diam x 9 in. long cylinder, observed with axis both vertical and horizontal; (c) 8-in. diam electric hot plate (laboratory model) with hot surface horizontal and operated at 3 different rates of heat input: low, moderate, high; (d) 12-in. diam x 5¼ in. high pan placed upside down on metal plate.

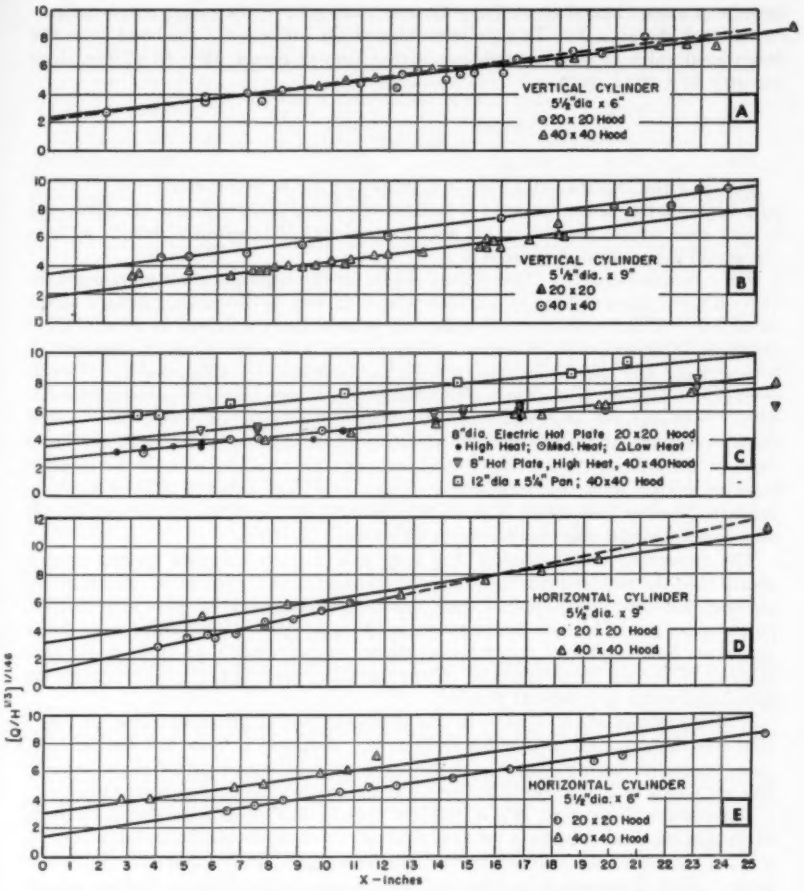


FIG. 3—PLOTS, FROM THE EXPERIMENTAL DATA, OF $[(Q/H^{1/3})^{1/1.46}]$ VS. X. A, B, AND C COVER FOUR SYMMETRICAL BODIES, WHILE D AND E COVER TWO ASSYMETRICAL BODIES REPRESENTED BY HORIZONTAL CYLINDERS

The method used here for determining Q is not an absolute one. Since the hot body, in most of the tests, had a circular cross section, the rising stream was also circular and it varied in diameter with distance above the body. Consequently, the stream did not correspond to the hood either in shape or size. Moreover, the rising stream has a high center-line velocity compared with its velocity at the outer boundary and this causes turbulence within the hood and encourages momentary escape of smoke at certain points around the edges of the hood despite the fact that the overall flow is adequate A better, although impractical arrangement would

have been a round hood of varying diameter, adjusted to the proper stream diameter at each distance X. To gain some idea of the influence of the hood upon the determined value of Q, tests were run with 2 sizes of hoods: 20 x 20 x 20 in. deep and 40 x 40 x 20 in. deep.

RESULTS

In Fig. 3 the experimental data are presented in the form: $(Q/H^{1/3})^{1/1.46}$ *vs.* X on arithmetic coordinate paper. A satisfactory linear relationship is demonstrated in each case with a positive value shown for the Q/H function where $X = 0$, as one would expect from Equation 6.

Q as a function of $H^{1/3}$. The plotted points for the laboratory hot plate operated at 3 different rates of heat input, shown at C in Fig. 3, obviously fall on a single trend line. The 3 rates of heat input were substantially different and, for a given distance, X, resulted in different values of Q. The ratio, $Q/H^{1/3}$, however, is constant for any particular distance X, as predicted by Equation 6. Further evidence in support of the Q *vs.* $H^{1/3}$ relationship is seen in the other experimental data. Although the electrical input to the test body was not deliberately varied, the convective heat output did vary from one test to another, owing to minor variations in air currents in the cubicle and because of the influence of the ammonium chloride smoke upon this component of the total heat output. Plotting $Q/H^{1/3}$ against X rather than Q *vs.* X definitely reduced the scatter and brought experimental points close together along the trend line.

The dependence of Q upon $H^{1/3}$ is a most important practical finding since it means that the required rate of exhaust ventilation for the capture of contaminated air over hot-process equipment does not increase remarkably with heat output. In practice, it will be difficult to estimate the convective heat output from many processes, but, in view of this relationship between Q and H, an error in the estimate of H will not be too serious. For example, a twofold error in H changes the calculated value of Q only 27 percent.

Q as a function of $(X + A)^{1.46}$: The demonstrated linear relationship in Fig. 3 confirms the exponential increase in Q as the distance, Z, from the hypothetical point source of heat, increases. If the exponent had a value other than 1.46, the plotted points in Fig. 3 would have followed curved rather than straight lines.

Agreement with constant, W_1: A constant appears in Equation 6 which is made up of certain physical constants for any given set of environmental conditions (g, C_p, ρ, T_a, C) but does not include any variables related to the dimensions of the heated body. With Q expressed in cubic feet per minute, heat output in Btu per minute and X in inches, as employed in the analysis of the data, this constant has a value of 0.198 and $K^{1/1.46} = 0.33$.

From the slopes of the lines in A, B, and C in Fig. 3, the four bodies with circular cross-sectional slopes, the values of $K^{1/1.46}$ and of K shown in Table 1 are obtained.

Since the 2 cylinders with horizontal axes presented asymmetrical shapes (D and E in Fig. 3), they were not expected to conform exactly to Sutton's relationship for a theoretical point source and are not included in the analysis.

The experimental values of K are smaller than the predicted value of 0.198. Error may reside in the experimental procedure (in which the shape and size of the exhaust hood was assumed to have no influence upon the Q *vs* X relationship), or in the calculation of H from Q and Δt. Another possible source of error is in the use of an improper value of C, the coefficient of eddy diffusion. Sutton's value of

TABLE 1—VALUES OF $K^{1/1.46}$ AND K FOR FOUR BODIES WITH CIRCULAR CROSS-SECTIONAL SLOPES

BODY	DIMENSIONS, IN.		$K^{1/1.46}$	K
	DIAM	HEIGHT		
Vertical cylinder	5½	6	0.24	0.124
Vertical cylinder	5½	9	0.25	0.132
Hot plate	8		0.20	0.100
Pan	12		0.20	0.100

$C = 0.17$ ft$^{1/8}$ was obtained from observation of the diffusion of chemical warfare smokes in the atmosphere over grass-covered plains. In the present experiments, in order for K to agree otherwise with W_1, the value of C would have to be 0.13 for the two 5½-in. diam vertical cylinders and 0.11 for the hot plate and 12-in. diam pan. Considering the greater stability of the atmosphere within the experimental cubicle compared with the general atmosphere, such decrease in C is not unreasonable.

Relation between A and diameter of heated body: The values of A, calculated from the experimental lines for the circular bodies in A, B and C in Fig. 3, are listed in Table 2 together with the predicted values, according to the relationship:

$$A = 2.15\ B^{1.14}$$

where

B = diameter of hot body.

The predicted value of A is based upon the assumption that the stream of rising air has an initial diameter equal to that of the hot body itself. This is not exactly true since there is a tendency for the rising stream to contract (somewhat like the vena contracta in fluid flow through an orifice) just above the top surface of the body.

There is fair agreement between the experimental and calculated values of A, the agreement falling between the values for the larger and smaller hoods. Un-

TABLE 2—VALUES OF A CALCULATED FROM EXPERIMENTAL DATA ON FOUR BODIES IN A, B, AND C OF FIG. 3

BODY	DIMENSIONS, IN.		VALUE OF A, IN.		
			FROM TESTS		PREDICTED
	DIAM	HEIGHT	SMALL HOOD	LARGE HOOD	
Vertical cylinder	5½	6	9.6	8.7	10.7
Vertical cylinder	5½	9	7.3	14.0	10.7
Hot plate	8		12.5	17.5	16.2
Pan	12		—	26.0	24.2

doubtedly, this reflects an error in determining the value of Q corresponding to different values of X and the influence of hood shape and size upon this relationship.

Effect of asymmetrical shape of hot body: Sutton's equation for free convection air flow, presented at the outset, is for a hypothetical point source of heat. The pattern of flow above such a point source would be symmetrical about the vertical axis and the rising stream would be circular in shape.

For a *line source* of heat of limited length, it is obvious that the Q *vs.* Z relationship will change as Z increases. At short distances relative to the length of the line, the flow pattern should approach that from a line source of infinite length; whereas at greater distances, the flow should tend toward that of a point source In view of these comments, there is little justification for presenting the experimental data in accordance with Equation 6, as has been done at D and E in Fig. 3. The limited findings are shown only to call attention to the fact that with asymmetrical heat sources, the analysis, based upon an assumed point source, can only be applied with caution. More experimental work is needed on convection flow over line sources to provide appropriate equations.

APPLICATION TO PRACTICAL DESIGN PROBLEM

Within the limits of this study, the experimental findings confirm the equation developed by Sutton, which relates the rate of air flow established over a heated body by free convection to the rate of convective heat output from the body and the distance above the body. It thus provides a theoretical basis for calculating the exhaust ventilation requirements for canopy hoods mounted axially within relatively short distances above heated process equipment having square and round shapes. The significant relationships are as follows (velocity, fpm; Q in cfm; H in Btu/hr; Z, X, A and B in ft):

$$\text{Axial velocity} = V = 9.6\,H^{1/3}\,(X + A)^{-0.29}$$

$$\text{Radius of rising stream} = r = 0.25\,(X + A)^{-0.86}$$

$$\text{Distance, } \textit{point source} \text{ to top of hot body} = A = 2.15\,B^{1.14}$$

$$\text{Rate of air flow} = Q = 1.95\,H^{1/3}\,(X + A)^{1.43}$$

There is an important practical limitation in the application of these relationships which can be of serious consequence. The equations are valid only in the absence of disturbing horizontal drafts which tend to break up the vertical mixing of the heated with the surrounding cooler air and also tend to tip the rising stream away from the vertical axis. It was necessary, in this study, to construct a tight, insulated cubicle with heat-reflecting inner walls, to eliminate such disturbances. Under practical industrial conditions, cross-drafts will always exist. Horizontal currents will vary in an unpredictable way and there appears to be no basis for introducing appropriate allowances for their effects in the equation. The problem, at the present time, has to be dealt with in an empirical fashion. The following steps for dealing with an actual installation are recommended by Hemeon[3]:

1. By observation, note the degree of tipping experienced by the rising stream caused by the cross-draft.

2. Make the canopy hood sufficiently large in its horizontal face dimension that it encompasses the rising stream when tipped to its maximum degree in any direction.

3. Calculate the theoretical quantity, Q, for the given distance, X (assuming $A = 2.15\,B^{1.14}$), and the area of the rising stream.

4. Calculate the design rate of exhaust ventilation through the hood as: $Q_d = Q + V_d\,(A_h - A_s)$, where Q_d is the design rate of ventilation; Q, the calculated rate of air

flow established by free convection; A_h, face area of hood, great enough to encompass rising stream when tipped to maximum degree; A_s, area of rising stream; V_d, arbitrary air velocity through the extra face area of the hood not occupied by the rising stream.

The design velocity, V_d, must be in excess of the velocity of cross-drafts to prevent the escape of contaminated air. Hemeon suggests that this may be as high as 150 fpm.

Example: Assume a round furnace, 4 ft in diameter and 4 ft high, with an average surface temperature of 800 F, ambient temperature 70 F. Assume for vertical surfaces, $h_c = 0.35\ (\Delta t/L)^{0.25}$ *and for the top horizontal surface,* $h_c = 0.48\ (\Delta t/L)^{0.25}$*, then convective heat output will be:*

1. Walls: $4 \times 4 \pi \times 0.35\ (730/4)^{0.25}\ (730) = 47{,}000$ Btu/hr.
2. Top: $4 \pi \times 0.48\ (730/4)^{0.25}\ (730) = 16{,}000$ Btu/hr.
3. Total convective heat output = 63,000 Btu/hr.

Canopy hood is to be located 8 ft above top of furnace. Observation reveals that, owing to cross-drafts, rising stream of air may be tipped from vertical as much as 10 deg. Determine diameter of hood face and minimum rate of exhaust ventilation.

Solution:

1. $A = 2.15 \times 4^{1.14} = 10.4$ ft.
2. $r = 0.25\ (18.4)^{0.88} = 3.3$ ft; $A_s = 34$ sq ft.
3. $V = 9.6\ (63{,}000)^{1/3}\ (18.4)^{-0.29} = 165$ fpm.
4. $Q = 1.95\ (63{,}000)^{1/3}\ (18.4)^{1.46} = 5500$ cfm.

To encompass the stream tipped 10 deg from vertical will require a hood with radius at the face of 6.5 ft; $A_h = 133$ sq ft.

5. Taking $V_d = 150$ fpm: $Q_d = 5500 + 150(133-34) = 5500 + 15{,}000 = 20{,}000$ cfm.

In this example, the effect of cross-drafts is seen to increase the total rate of exhaust ventilation 3.5 times over the basic rate of flow in the rising stream of heated air. Thus, the disturbing air movement in the surrounding space is the overwhelming factor which determines the rate of exhaust ventilation. Failure to recognize this has been the cause of poor performance observed in many industrial installations of canopy hoods over hot-process equipment.

PRACTICAL MEASURES TO AID HOOD PERFORMANCE

Measures to improve the performance of canopy hoods over the hot process equipment, to avoid the disturbing effects of cross-drafts, should be in the direction of eliminating the basic cause of difficulty. Increasing the size of the hood and the rate of air flow through it are second steps in the solution of the problem. Obviously, the first objective should be to locate the hood as close as possible to the process equipment. One might hang flexible curtains from a fixed hood to permit entrance of other equipment to the space between the process and fixed hood or support the hood itself with a counter-balanced device and flexible pipe connection to the exhaust main so that it can be positioned close to the equipment during most of the operating cycle and easily raised when operations require more space above the equipment. Partial enclosures around the equipment may be employed in some cases to shield the operation from cross-drafts. Sources of disturbing drafts should be identified and eliminated so far as possible. The use of a man-cooling fan to offset radiant heat exposure around hot equipment is one source of disturbing cross-drafts which upset exhaust hood performance. Radiation shielding is a more basic method for control of such heat exposures and should be

employed to the fullest degree. Supply air for local control ventilation and cooling should be introduced in a manner to avoid disturbing the exhaust hood performance. The air blast from a local heating unit is another potential source of disturbance which can be avoided by careful planning. Nearby equipment with moving parts may create air motion which upsets hood performance. Suitably placed baffles will often correct such situations. Location of the exhaust ventilated equipment near windows, doors or other air intake openings is a common source of trouble which should be avoided.

SUMMARY

1. On a laboratory scale, Sutton's equations for predicting the radius, velocity and total rate of flow in the stream of rising air over hot bodies, at various distances above the body, have been confirmed. The rate of flow is proportional to the one-third power of the convective heat output from the heated body and increases with the 1.46 power of the distance above the theoretical *point source* of heat. The *point source* is located at a distance below the top surface of the body which is proportional to the 1.14 power of the diameter of the hot body.

2. Effect of cross-drafts in the surrounding space is to tip the rising stream from the vertical. The exhaust hood must be large enough to encompass the rising stream when tipped maximally in any direction and an extra quantity of air must be drawn through the remaining area, not supplied by the rising stream, to ensure complete control. In a practical situation, this added requirement may increase the overall ventilation rate several times over the basic rate of flow in the rising stream.

3. There are at present no recorded experiences in support of or contradictory to the suggested design procedure. Some actual field studies in a variety of industrial situations are needed to confirm the practical usefulness of Sutton's equation and Hemeon's empirical corrections for the effects of cross-drafts.

REFERENCES

1. Turbulente Ausbreitung eines Stromes Erjitzter Luft, by W. Schmidt (*Zeit. Angew. fur Math. und Mech.*, Vol. 21, 1941, p. 265).
2. The Dispersion of Hot Gases in the Atmosphere, by O. G. Sutton (*Journal Meteorology*, Vol. 7, 1950, pp. 307–312).
3. *Plant and Process Ventilation*, by W. C. L. Hemeon (Industrial Press, New York).

DISCUSSION

W. C. L. HEMEON, Pittsburgh, Penn. (WRITTEN): Sutton's development contemplated the application of this equation to the behavior of hot stack gases in the open atmosphere and involved some assumptions from other experimental work in quite different circumstances. When we adapted his equations in the form shown by the authors, for application to High Canopy Hood problems we had, naturally, some reservations in the absence of experimental confirmation. The authors have now confirmed their validity and this is gratifying to us.

In the same chapter of our book* another analysis was presented of convection air flow based on considerations of velocity head for application to low canopy hoods. This

* Hemeon, W. C. L. *Plant and Process Ventilation*, Chapter 8., The Industrial Press, New York (1955).

led to the conclusion that the vertical flow of heated air at the top boundary of the hot object must be related to the dimensions of the object, *i.e.*, its projected horizontal area, A_p, and its vertical dimension, m, as follows:

$$q_0 = 29H^{1/3} A_p^{2/3} m^{1/3}$$

When this equation was applied to the Hatch and Barron-Oronzco cylinder 5½ in. x 6 in. we obtained

$$q_0 = 6.8\ H^{1/3}$$

This can be compared with our turbulent mixing formula, the one treated in the present paper, by eliminating the term Z, *i.e.*, $A + X$. This is done for the same test object using the geometric equation $A = 2.15\ B^{1.14}$. When the substitution is made we obtain

$$q_0 = 6.3\ H^{1/3}$$

The close agreement is particularly striking in view of the basic difference in approach employed in each development. It seems reasonable to believe that for the condition where Z has a value of zero, the agreement is coincidental and that the former of the 2 equations above is the more appropriate one. It would be interesting to study its application to some of the other data presented by the authors.

P. J. MARSCHALL, North Chicago, Ill. (WRITTEN): As a member of the TAC on Industrial Environment, I wish to thank Professor Hatch, a fellow member of this committee, and Professor Barron-Oronzco for the time and effort they have so generously contributed in this investigation and report.

I particularly like their practical suggestions on what can be done to improve the performance of hoods.

In view of the present trend to provide more comfortable working conditions in industrial plants having hot manufacturing processes, and comfort air conditioning in research laboratories where hoods for the control of fumes, gases and heat are very common, it is very important to pay more attention to the basic cause of poor performance of canopy hoods.

GEORGE M. HAMA†, Detroit, Mich. (WRITTEN): The authors are to be complimented for the excellent presentation of design data for canopy hoods over hot processes. They have taken the equation of Sutton, which applies to the discharge of hot gases from stacks to the atmosphere, and used it in conditions of relatively short distances for hot processes under canopy hoods. Their experimental data indicates that hot gases from industrial processes basically follows the equation, with the exception of a slight discrepancy in the constant K.

The presently used design data, most frequently applied to canopy hoods, treats the canopy hoods as a booth with 4 sides open, and bases the air flow on the sq ft of open area between the hood perimeter and the floor. This approach makes no allowance for the thermal effects which tend to cause the hot gases to rise vertically into the hood and which entrains room air, greatly increasing the volume of gases arriving at the hood opening. We have all seen hoods into which the smoke from a hot process flows into the hood and then curls out at the edges because the volume of air exhausted from the canopy is too small. The design data given in the paper of Hatch and Barron-Oronzco appears to give a much better estimate of air volumes from hot processes. The paper also confirms an opinion I have had for some time, and that is that open canopy hoods should be generally applied only to hot processes.

Important points to be noted from the paper are that the air volumes to be exhausted through the canopy increases considerably with the diameter or size of the hot process

† Chairman, Committee on Industrial Ventilation, American Conference of Government Industrial Hygienists.

and with the distance from the canopy to the process, but only moderately (as the cube root) with the heat output (Btu) of the process.

The authors do not state what the optimum size is for the canopy hood. There appears to be a need for data on the optimum cross sectional opening and optimum depth of the hood. I would also like to see some confirming experimental data on the material suggested for making allowance for the cross-drafts *tipping* of the hot gas plume from its vertical axis. In the example cited a 4 ft diameter furnace, 8 ft from a canopy hood, requires 5500 cfm in still air with no cross drafts. Under assumed conditions of cross drafts, which it would appear might be frequently expectable in practice, 20500 cfm would be required. In considering some canopy hood installations which I have seen on similar hot processes, I wonder if this 20500 cfm air volume isn't a little high. It appears that for the design of hoods which are to be installed on new processes or those not previously observed by the designer, we need data on normally encountered cross drafts and the degree of tipping of hot gas plumes.

It is not a reflection on the excellent presentation of the authors but it is unfortunate the design equations involve decimal and fractional exponents which require logarithms or a log-log slide rule to resolve. This excludes their use by most of the practical designers and installers of industrial ventilation equipment. In the *Industrial Ventilation Manual of the American Conference of Governmental Industrial Hygienists* to meet the need of the practical designer, we attempt to present nothing more difficult to resolve than a few square roots. It is possible, however, that the design data of the excellent paper of Hatch and Barron-Oronzco can be presented graphically so it can be readily used and understood by all.

BENJAMIN FEINER, SR.‡, New York, N. Y. (WRITTEN): For a long time there has been a need for basic information on the behavior of air above hot processes, both free convective flow and secondary induced air, in order to permit more rational design of local exhaust ventilation. This paper, as well as Hemeon's earlier paper, constitute an important beginning to the solution of the problem.

An interesting corollary to this paper would be, as the authors suggest, to institute field studies to determine the validity of the equations. Such field studies could also be very useful to the designer if they could set up tables of predicted cross-drafts in a large variety of workrooms and convective heat outputs for a large variety of actual hot-processes, tanks, and similar installations. Most people engaged in design would not have ready access to such data and would find compilation, even if only approximate, very useful.

Several avenues of study present themselves. In many hot operations, canopy hoods are not desirable or possible because of considerations such as the passage of cranes, or because the extreme toxicity of the contaminant or the method of operation would place the worker's breathing zone in the path of contaminant flow. In such cases, rear exhaust in the form of lateral slots, rear hoods, partially overhanging hoods or similar methods would be required. Can the equations be modified for rear exhaust?

Another point of interest is the application of the equations to design of natural draft ventilation (assuming minimum cross-drafts and 100 percent positive make-up air supply). Theoretically, there appears to be no reason why this should not be possible using properly sized and located hoods and ducts.

The effect of hood design and location would also seem to be important. In the writer's own experience, distance above the source of heat, hood base area, angle of transformation, flanging and baffling are critical factors almost equally as important as ventilation rate. An improperly designed and located hood will usually not prevent spill of expanding air under its periphery, even with very high ventilation rates. Conversely, refinements to the standard unadorned canopy hood should reduce the cfm obtained by the equations. A quantitative study of the effect of these refinements would be useful.

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‡ Industrial Hygiene Engineer, New York Department of Labor.

LINN HELANDER, Manhattan, Kans.: Was the pressure in the hood (Fig. 2) measured?

R. C. LAST, Melbourne, Australia: Professor Hatch, I think your contribution to the realm of industrial exhaust is a very valuable one. In looking over the paper and listening to your lecture, I have a few questions to ask on the experimentation procedures.

First, in reference to your heated bodies and the measurement of heat flow, you considered several shapes and sizes of heated bodies. To get the best evaluation of convection, why did you not consider a heated body with a plain surface rather than a solid surface? Also in considering a flat surface, you will have to reduce the side convection current from the hot surface on the side. I mention this because I think that the flat surface is the only applicable one to apply to a point source.

Second, with the heated body and in reference to the room, you mention that the room is lined with aluminum finish. Why not then have the heated body with its sides also aluminum coated so as to reduce side radiation. Are you only considering direct radiation and convection from the top surface?

Third, your hood sizes are rectangular. If we are considering a cone, with an equivalent point source, why was the hood not made circular?

Fourth, in the measurement of air flow we are measuring free convection. I can see from the diagram that flow has been assisted by a fan. Do we assume that the fan was first balanced without the use of the hot body, and then air flow increased? Is this what we are measuring, when we talk about Q, or are we talking about the total air quantity?

AUTHORS' CLOSURE, (Mr. Hatch): I should like to thank all of these gentlemen for their very valuable comments on our paper and for their contributions to further consideration of the problem.

In respect to Mr. Hemeon's remarks, I should like to say it was a bit frustrating to carry on a piece of laboratory work while I had before me his excellent book in which he had treated the subject so adequately. In effect, all that we did was to confirm what he had already published.

Perhaps all may not have followed altogether the point he was making as to a different approach to the problem. Using the published experimental data, of which there is a considerable volume concerning the characteristics of convection flow around heated bodies both with respect to temperature and velocity, he arrived at a value of Q_0 which, as he said, is obviously a function of the shape and size of the body. He derived from the data an equation in which the convection flow is proportional to the heat output to the one-third power and having a constant which agrees very well with the constant in Sutton's equation and with the constants which emerged from our work. Sutton dealt with the problem flow above a hypothetical point source of heat. Mr. Hemeon's analysis, in which he gives full recognition to the size and shape of the body, appears to be in quite good agreement.

We used, in our equations, the exponents as they emerged from Sutton's work. The odd exponents of 1.14, 1.46 for the distance and so on, are not empirical. They are derived from the theoretical relationships and enter in some degree into the dimensional satisfaction of the relationship. For practical purposes, however, Mr. Hemeon has substituted for 1.46 the rounded value of 3/2, and for the exponent 1.14 he uses 1.0. These exponents can be handled on a slide rule and do not introduce serious error.

Mr. Helander asked if we measured the pressure just in front of the orifice. We did not in this work. Earlier we attempted to characterize the rising stream by direct measurement of velocities and temperatures in the stream, but, owing to the unsteady flow we could not obtain check readings. There was just too much inherent instability in the stream.

Mr. Last asked why we did not work primarily with heated surfaces rather than with bodies having depth. We did, as a matter of fact, work with both. We used bodies of cylindrical shapes having various depths and also simple plane surfaces. His question is: does the geometry of depth affect the problem? I can only say that within the limits

of our experimental approach, the plane surface and cylinder appeared to agree and both seemed to fit the point source concept very well.

We also studied bodies other than simply circular shapes. We studied rectangular shapes and long and narrow shapes and discovered, as one would suspect, that with rectangular shapes, one has a situation somewhere between a point source and a line source. At the time we had no mathematical development for line sources that would permit us to examine the data on a rational basis. There is need for the development of such a relationship because we have many situations where the characteristic geometry is one of length rather than a circular one.

Mr. Last suggested we might have coated the hot body with aluminum to eliminate heat loss by radiation. We did it, only to have our bodies burn up. They heated up to a red heat and then the heating elements disintegrated. To stay within practical limits in the laboratory we had to abandon the aluminum coating and resort to the aluminum lined chamber.

We used rectangular hoods of two different sizes in order to note the extent to which the hood size and shape influenced the results. Unquestionably they did introduce error. Ideally, we should have used a round hood and it would have been proper to vary the diameter as we moved the body up and down to keep the diameter of the hood in agreement with the diameter of the rising stream. Hence, the measured value of Q is questionable. It is Q_x, the quantity of air coming up, plus whatever else the hood itself is contributing to the total. I can only say that when we used this value we came out with constants that were not greatly different from those of Sutton.

The fact that we were exhausting air through the hood raises a proper question as to the degree to which the mechanical withdrawal of air increased the value of Q. I think not very much. The hood, as Hemeon would term it, was behaving as a receiving hood rather than an exhaust hood. The requirement of the hood was being satisfied by the rising stream. This would not be true if one were to use a much larger hood relative to the dimensions of the rising stream.

Mr. Hama's comments are most welcome. He calls particular attention to the disturbing influence of cross drafts and urges the need for systematic study of this aspect of the problem in actual industrial situations. He emphasizes the point made in our paper that the exhaust requirement to offset the effect of cross drafts may be the controlling factor in design rather than the quantity of air in the rising stream. I quite agree.

We subscribe also to his desire to reduce the equation to simpler form and, perhaps, present the essential information in graphical form for easy use by design engineers.

Mr. Feiner, who was an old associate of mine years ago and has had experience in wrestling with industrial ventilation problems, made some good points. Our work was done in the laboratory with a very small body. The highest heat input we used was 1,500 watts, which is very small, indeed. Obviously this laboratory work has to be checked with studies on actual process equipment as we encounter it in industry.

Mr. Feiner suggests the possibility of applying the equations to the performance of a hood employing natural draft. I see no reason why this cannot be done.

He points out, as well, the need now for getting data on the performance on lateral hoods, and this is indeed the subject of further work which we have in progress. We need such data a great deal more, in fact, than we need the information on canopy hoods.

He suggests that the design of the canopy hood with respect to the degree of depth and flare is important. There is no question about that. We have seen in industry that shallow hoods that are relatively abrupt do not perform as well as the others. Most of the cases of failure, however, are the result of this fact: the hood is exhausting less air than is being delivered to it by the hot process.

1603

No. 1603

RADIOACTIVE PROCESS VENTILATION

By S. H. GLASSMIRE* AND J. P. WAHLEN**, LOS ALAMOS, N. M.

THE ADVENT of atomic energy brought forth a new industrial or occupational health hazard:—that of exposing man to the effects of ionizing radiation; that is, any radiation which is capable of producing ions when passing through matter. Working with radioactive material involves the possibility of exposing operating personnel to alpha, beta, gamma, and even neutron rays that are constantly emitted by radioactive substances. These rays may enter the body either externally by absorption through the skin or internally by inhalation or ingestion. Once they have entered the body, the effect of nuclear radiation is the destruction of various molecules that play an important part in the function of living cells.

Generally speaking, exposure to nuclear radiation is minimized in two ways: by shielding and by ventilation. *Shielding* is used to reduce exposure by radiation from an external source or one outside the body. Shielding is designed to reduce the higher energy gamma ray intensity to near zero at the human occupancy area. If gamma ray penetration is sufficiently reduced, the other lower energy or less penetrating, particles will be of no concern.

Proper *ventilation* is the prime precaution against internal exposure to radiation originating within the body. Dust in the working atmosphere surrounding nuclear activity has dangerous induced activity or particulate effluent of the nuclear material itself may contaminate the air. Once such radioactivity has been inhaled or ingested, it is in a position to do greater damage since it is closer to body tissue and subject only to depletion by decay or biological elimination processes.

There is no bodily sensation to indicate the presence of nuclear radiation and instruments must be used to determine where hazard exists. Since instrument warnings cannot be constantly located everywhere within the human occupancy areas the ventilation must be proper and adequate at all times.

Buildings, laboratories, and enclosures containing or involving handling nuclear material within their confines obviously require unique and exacting ventilation, filtration, and air distribution considerations.

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Presented at the 63rd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Chicago, February 1957.

VENTILATION REQUIREMENTS OF A NUCLEAR LABORATORY

Requirements for various process laboratories may differ slightly. However, the following considerations may be set forth as pertinent and typical requirements for nuclear laboratories in general:

1. Adequate quantities of clean, treated, 100 percent fresh make-up air must be delivered to the laboratory areas. Air quantity requirements for contaminated human occupancy areas vary on individual jobs ranging from perhaps 10 to 90 complete air changes per hour. Much of the work on radioactive materials is analytical in nature requiring filtering and treating of the inlet air. The inlet air is 100 percent fresh since all exhaust, even though filtered, is potentially hazardous.

2. Adequate by-passes, crossovers, and auxiliaries must be provided so that failure or maintenance shutdown will not leave the laboratory without ventilation.

3. Exhaust air quantities must be sufficient to maintain adequate face capture velocities (100 to 250 fpm minimum average) across open faces of hoods and across equipment so as to assure carriage of contaminants away from operating personnel. (80 fpm is the absolute minimum at any one point.)

4. Exhaust duct velocities must be sufficient, generally 2000 to 5000 fpm, to suspend and carry heavy particulate contaminants to the filters.

5. Exhaust air filters must be:

a. Adaptable so that the collected particulate matter on used filters can be recovered.

b. Highly efficient on submicron particles.

CMR BUILDING—LOS ALAMOS SCIENTIFIC LABORATORY

General Remarks: The CMR Building is an example of a laboratory embodying these special considerations.

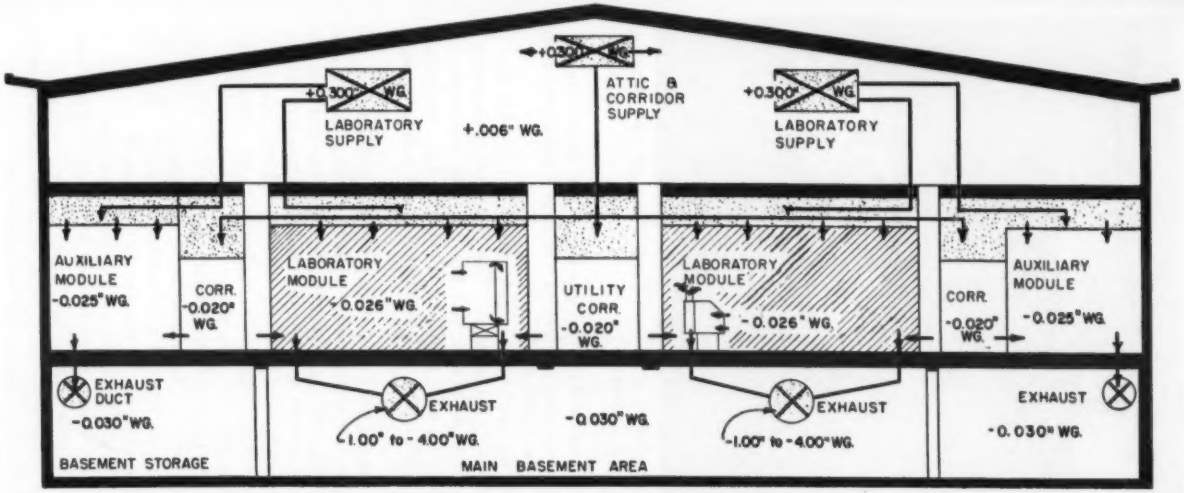
It is essentially a 3-story structure with a full attic and basement which are auxiliary to the main floor which contains the radioactive chemistry and metallurgical laboratory modules. It is a winged structure with 5 radioactive laboratory wings, each 267 ft long, branching off a 650 ft long spinal corridor. (See insert plan on Fig. 1.) Construction was completed in 1952 at a contract cost, excluding special equipment, of between 11 and 12 million dollars.

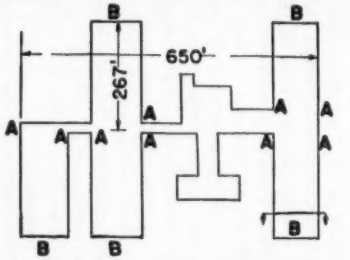
General Air Distribution and Flow Pattern: Fig. 1 is a cross-section of a typical wing showing the general flow pattern. It is interesting to note the proportional amount of auxiliary space which is subsidiary to the main operation modules shown in cross hatch. Air enters the building in the attic over the spinal corridor of each wing at positions marked *A* on the plan insert of Fig. 1. Air is exhausted at the ends of each wing at positions marked *B*.

Supply air is distributed to various laboratories from mains in the attic to chambers above perforated metal ceilings in each module. The air is evenly distributed through perforations in the ceiling and is exhausted through equipment or by grilles at the floor into basement mains.

In general the air flow direction is from *cold* (least radioactive) to *hot* (more contaminated) areas. This flow pattern is maintained by carefully controlled static pressure regulators controlling the supply and exhaust to and from the various areas. The static pressure controllers, with sensing elements located in the main, regulate flow by automatic regulation of vortex type dampers. Fig. 1 shows the theoretical pressures maintained in the various areas in order to channel the flow in the proper direction from *hot* to *cold* areas. All pressures indicated are referred to atmosphere taken at 0.000 in. water. As can be seen, the static pressure gets less progressive (more negative) from cold to hot areas.

Each wing is currently handling approximately 80,000 cubic feet of air per minute or 400,000 cfm is handled by the whole building. Proper functioning of all ven-





PLAN - GMR BUILDING
LOS ALAMOS SCIENTIFIC LABORATORY

FIG. 1—TYPICAL CROSS-SECTION OF WING. THE GENERAL PATTERN OF AIR FLOW IS FROM THE ATTIC TO THE BASEMENT AND ON THE OPERATING FLOORS FROM CLEAN AREAS TO POTENTIALLY RADIOACTIVE CONTAMINATED AREAS. AIR ENTERS THE BUILDING THROUGH THE ATTIC AND IS EXHAUSTED FROM LABORATORIES AT THE FLOOR AND IS THEN CARRIED TO THE AIR CLEANING TOWER AT THE END OF THE WING. FOR INTERIOR PRESSURES SHOWN THE ATMOSPHERIC PRESSURE OUTSIDE OF BUILDING IS ASSUMED AT 0.000 IN. WATER

FIG. 2—VIEW OF CENTRAL BOARD WITH LIGHTS FOR EACH PIECE OF VENTILATION EQUIPMENT

tilation equipment is constantly monitored in the Central Control Room (see Fig. 2). Any malfunctioning of any ventilation equipment will flash a red light on the panel which gives the location of the faulty equipment.

Supply System: The supply air for each wing is motivated by 3 *NAFM* No. *Q* radial bladed centrifugal fans with 25 hp motors. The 3 blowers discharge air into each of the 3 large supply mains in the attic (see Fig. 1) which essentially act as low pressure (+0.02 in. to +0.40 in. w.g.) plenums. From the mains the supply air is distributed via branches to chambers above the perforated ceilings in each laboratory module and corridor.

Each of the 3 supply systems in each wing draws 100 percent supply air from the atmosphere across screened louvers at an attic level 30 to 40 ft off the ground. The air is first drawn across oil impregnated wire roughing filters—thence across the preheat bank of steam heating coils set at 55 F—thence through the spray washer followed by moisture elimination plates—thence through the reheat coils set at 64 F—thence across a final set of oil impregnated wire filters—thence through the vortex damper through the blower into the attic mains.

Moisture pickup and thus relative humidity can and has been partially controlled by regulating the spray head pressure. This is done by modulating the valves of the spray headers according to the dictates of a humidistat located in the after spray chambers. Humidity can be partially controlled in this manner as long as the outside relative humidity is below the set level of the humidistat. This is true most of the time in Los Alamos with a mean relative humidity level of about 20 percent.

The spray water for the washers is recirculated. Build up of solid concentration is minimized by continuous blowdown at a rate equal to evaporation.

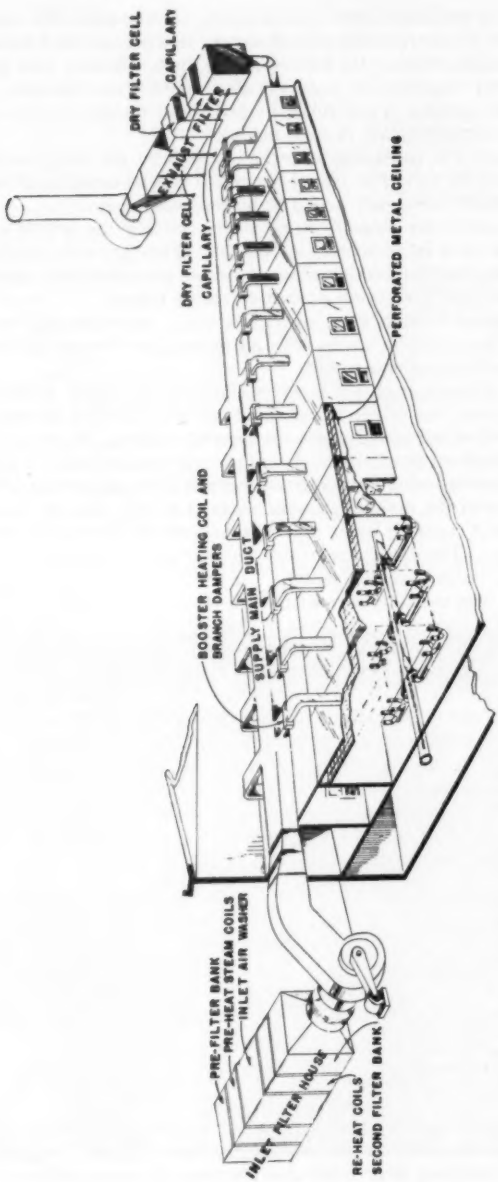


FIG. 3—PICTORIAL SECTION OF HALF OF TYPICAL WING SHOWING VENTILATION SYSTEM

Supply amounts and main duct static pressure are automatically regulated by a vortex damper on the suction side of each supply blower controlled according to the dictates of the static pressure tip located in the duct. Branch duct supply to individual modules is regulated by hand-operated louver type dampers. Final heat to the individual modules is controlled by hot water booster coils in each branch duct regulated by thermostats in each module.

Exhaust System: The contaminated exhaust air from the main modules in each wing is exhausted by 2 *NAFM* No. *Q* backward curved centrifugal fans handling approximately 38,000 cfm each with 100 hp motors.

The exhaust mains are smooth round ducts running the length of each wing tapering from 18 to 48 in. diameter (see Fig. 3). They are constructed of 3/16-in. type 316 corrosion resistant stainless steel and are provided with automatic spray washers for additional protection against corrosive fumes.

Specially designed *horses* or *dogs* (see Figs. 4 and 5) act as branch ducts to supply the main. The legs of these horses draw air at the floor through grilles or through hoods or other laboratory equipment.

Desired flow through each leg is maintained by an orifice plate (see Fig. 4). Pressure drop across the orifice is measured and the quantity is then ascertained from curves plotted by actual cfm experimental values. None of the standard formulae for calculated *Q* are used. Each horse is equipped with a positive acting 16-in. wafer butterfly valve for additional branch flow adjustment (Fig. 4).

Static pressure in the main duct, which acts as a high velocity header, is maintained at a desired constant by an automatic vortex damper on the suction side of each exhaust fan. The damper regulates according to the dictates of a static pressure tip located in the exhaust mains.

Static regain does not play a great part in the overall system which is contrary to what might be expected. Changes in duct velocities are purposely kept small so as to keep the conveying capacity satisfactory. In fact, regain is sacrificed by utilizing converging stacks to give a greater throw and dispersion into the atmosphere.

Exhaust Air Filters: All discharge air from laboratories handling radioactive materials must be considered as contaminated and must be filtered to low tolerance levels before release to the atmosphere. Just how clean the discharge effluent must be is as yet unanswered.

Type of air filtration varies according to the nature of the radioactive contaminants. Many different types of air cleaning equipment are and have been used in the AEC industry. These include:

A. Settling chambers
B. Centrifugal separators
   1. cyclones
   2. baffle chambers
   3. Venturis
C. Wet filters
   1. air washers and spray towers
   2. wet cells (capillaries)
   3. combinations of 1 and 2
D. Electrostatic precipitators
E. Impactors
F. Dry filters
   1. viscous impregment filters
   2. dry filters
   3. high efficiency filters

The most satisfactory results are obtained by using a combination of some of these collectors in series. Various kinds of filters are arranged in order of increasing efficiency with roughing filters preceding the ones of greater efficiency.

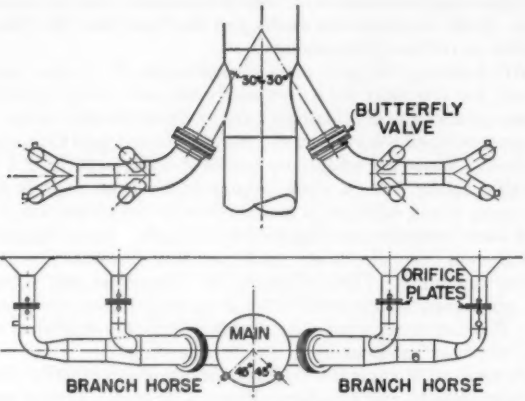


FIG. 4—SPECIALLY DESIGNED HORSES OR DOGS WHICH ACT AS BRANCH DUCTS TO SUPPLY EXHAUST MAINS. NOTE PRESENCE OF ORIFICE PLATES AND BUTTERFLY VALVES TO REGULATE FLOW IN EACH BRANCH

Replacement or cleaning filters obviously involves exposure to radiation and poses a special problem.

FIG. 5—VIEW OF EXHAUST MAIN AND BRANCHES IN A TYPICAL WING

The particulate matter collected on used filters often times is valuable and must be recovered. Both the collecting media and the form that the filters are in must lend themselves to recovery operations.

In the CMR Building the particulate contaminants discharged into the exhaust air stream are, for the most part, irregularly shaped, heavy metallics. Particle size varies but most of the contaminants are in the submicron range.

The exhaust from each wing enters the filter tower and goes through the capillary air washers first. These are essentially banks of 20 in. x 20 in. x 8 in. thick cells with coarse glass (about 0.01 in. diam.) filaments set at an angle to the air stream. Water from spray heads, oriented in the direction of the air stream, is sprayed over the cells and water cascades over the entire cell bank. Such capillaries are highly efficient removers of corrosive fumes, acid mist, and chemical vapors. This is one of their prime functions. Their efficiency on submicron particulate matter has never been ascertained at this installation since no samples are taken directly behind them. *Zig-Zag* water eliminator plates follow the capillaries to protect the downstream dry-filter pads against water saturation.

Next in the series of filters is the first bank of dry filters utilizing glass fiber media depending upon density and roughing efficiency desired. Coarse spun glass pads are often used depending upon the size of the particulate matter.

These first dry filters are followed by another wet cell identical to the first one. This second bank of wet cells is used only under certain process conditions of the effluent.

Final cleanup is accomplished by commercial glass fiber mat filters or by specially compacted spun glass pads of varying density manufactured to Laboratory specifications.

Using various combinations of these media in the series of 5 filters (3 wet and 2 dry), overall efficiencies in excess of 99 percent are maintained.

Local filtering is employed at the source of exceptional *hot spots*. Efficiencies checked under actual operating conditions run about 99.99 overall and 99.95 for particles smaller than 0.25 microns. (All efficiencies are measured by drawing off equal air samples at the same rate before and after the filters. The air is drawn across millipore media which are counted for total radioactivity. The inlet and outlet are then compared.) Local filtering at the source of known hot spots avoids undue contamination of the central filtration units.

Radioactive *hot* cells always employ local exhaust filtering right at the wall of the cell. Cells are essentially rooms or chambers with excessive radioactivity and generally require walls of cement or other material several feet thick to shield gamma radiation. High air flow through such enclosures is generally desired. Exhaust filtering is generally performed at the walls or the floor of the cell. The filtering media must serve a most difficult dual capacity as a shield and a high efficiency particulate filter or shielding must be built around the hole. At Los Alamos we have experimented with a type of sand filter for such purposes. Volcanic tuff, a highly vesicular consolidated volcanic ash, is ground to size of 1 in. down to 50 mesh. A thick bed of such material acts as a good gamma radiation shield. Experimental tests give overall efficiencies over 99 percent at about 1 in. water pressure drop. (See Fig. 6.)

OTHER NUCLEAR VENTILATION APPLICATIONS

Reactors: Most reactors to date are experimental types which must waste heat, nuclear phenomenon being of first interest. In order to maintain high neutron flux

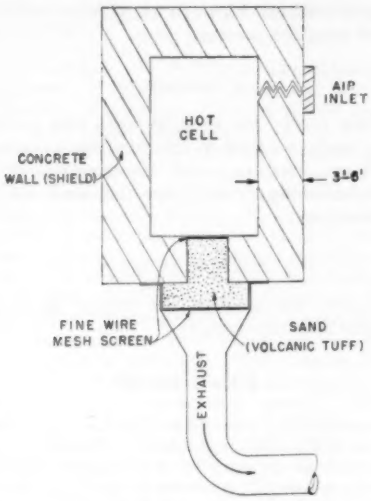


FIG. 6—DIAGRAM SHOWING HOT CELL WHERE HIGHLY RADIOACTIVE MATERIAL IS PRESENT WITH SAND FILTER IN EXHAUST UNDER FLOOR

levels, the associated thermal energy must be dissipated lest the reactor temperature exceed the materials' melting points.

The most common coolant is air, whose low specific heat makes the quantities involved very great. A reasonable figure for air flows is 25 cfm per kilowatt. The inlet air must be filtered by *highly efficient filters* to remove the likelihood of atmospheric dusts becoming radioactive by neutron bombardment. The exit air must be absolutely filtered, also, to remove radioactive dusts originating from within the core. As a rule, the coolant air passes through small diameter, long passages, integral with the reactor. The friction loss is large and the temperature of the exhaust air is quite high, around 300 F. At Brookhaven, the air is discharged through 320-ft stacks and requires enormous fan horsepowers.

Power Engines: Since power plant reactors have the purpose of being heat sources, air cooling should not be used. Other media will transfer the thermal energy to the heat exchanger. Ventilation required will be for reducing health hazard to operating personnel and not for controlling reactor temperatures.

Power engines for ship or aircraft using nuclear fuel propulsion utilize the heat and discharge the exhaust. Dispersion and dilution by the atmosphere or the sea is currently the only safeguard against contaminants contained in the exhaust. Protection for the crew is maintained by progressively greater negative pressures assuring flow direction from cold to progressively more contaminated zones.

Nuclear Bomb Shelters: Nuclear weapons have and will necessitate the need for shelters that afford protection against radioactive fallout. This involves a ven-

tilation problem—again utilizing high efficiency filters with blowers and motors sufficient to pull their required pressure drop.

CONCLUSION

The advent of atomic energy has brought forth new problems to confront the ventilation engineer. Such problems to date have been mainly limited to classified government installations. However, with commercial and peacetime utilization of atomic energy now becoming a reality, the problems and special considerations will be general with industry.

ACKNOWLEDGMENT

The illustrations used in the paper are the work of Joyce B. Engle to whom the authors express grateful acknowledgment.

DISCUSSION

P. M. ENGLE†, Syracuse, N. Y. (WRITTEN): The paper is a good summary of what is at best a very extensive and complicated subject. Because of our position in the high efficiency air filtration field, we are sometimes called upon to assist in the design or check the plans for ventilation systems of industrial facilities for the handling of radioactive materials. We are glad to do this, but we find that it is often made more difficult by an uncertainty as to the amount and types of radioactive materials to be handled.

Usually, the engineer must select equipment capable of meeting severe radioactive conditions if there is some reasonable chance that they will be encountered. Extensive changes in a ventilation system after it has become contaminated are always difficult and expensive and may be completely impractical.

Here, I will confine the remainder of my remarks to my major interest, air filters. I started to say the exhaust air filters, but first want to mention that high efficiency air filters on the supply system are also important. This applies not only to reactors as mentioned by Mr. Wahlen but also to laboratories. In the laboratory the ordinary dust in the supply air may not be activated but most of it will eventually end up on the exhaust filters. It is much easier and more economical to remove this material on uncontaminated filters than to have it pass thru and shorten the life of the exhaust filters. Really clean air in the laboratory itself is a valuable by-product of this arrangement.

In regard to the exhaust filters, we feel that the paper does not emphasize sufficiently the extensive use of extremely high efficiency type filters as the final clean-up for contaminated exhausts. These are the filters which test 99.95% efficient on 0.3 micron di-octyl-phthalate smoke. This situation is probably due to the fact that for a number of years these filters have been in such common use that they are becoming *old hat*. I believe the authors will agree, however, that at most of the AEC sites these extremely high efficiency type (also known as AEC or CWS-6 type) filters are the standard for radioactive exhaust applications and that other means are employed only for special conditions.

For example, lower efficiency fine fiber glass media filters which still have at least the efficiency of electrostatic precipitators are being employed at Los Alamos for exhaust filtration in a part of the CMR building described in Mr. Wahlen's paper. However, the extremely high efficiency type filters are employed for the most critical of the contaminated final exhaust filtration installations at that site.

Recent developments have made the extremely high efficiency filters available in a number of new materials of construction. These new models which provide almost any

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† Cambridge Filter Corporation.

desired degree of corrosion, temperature, moisture, and fire resistance are making these filters even more popular.

C. R. WHERRITT‡, Los Alamos, N. M. (WRITTEN): The authors are to be congratulated upon the excellent description of the CMR Building ventilation and exhaust system at Los Alamos, New Mexico. The review of the ventilation problems encountered in atomic energy research work is well presented.

I should like to point out some operating experiences with the exhaust ventilation system of the CMR Building with respect to capillary washers in particular. As the authors have pointed out in their discussion, such capillaries are highly efficient removers of corrosive fumes, acid mist, and chemical vapors. However, in some of the wings in the CMR Building, the problem of maintenance of these capillary washers became so acute that certain changes were necessary from an economic standpoint.

Although the domestic water available in Los Alamos cannot be called hard by most standards, the amount of solids is high enough on the average, (150 to 190 ppm total solids with 30 to 40 ppm calcium and magnesium) that appreciable amounts of mineral deposits were formed on the capillary washers almost as soon as they were installed. Since these filters were contaminated with radioactive particles, it was difficult to maintain them because of the hazard of exposure to the maintenance people and the undesirable shutdown required. It soon became apparent that this problem would become increasingly more difficult as the solids continued to build up on these washers. Two engineering studies were undertaken to eliminate this problem. The first study considered the use of demineralized or distilled water in the air washers to avoid the mineral deposits being formed. The other was to replace the washers with dry type filters.

Since the concentration of corrosive fumes and vapors in the exhaust mains appeared to be lower than had previously been anticipated, it was decided that a dry type filter arrangement might be more economical and easier to maintain.

In one of the exhaust filter systems where the capillary washers had become extremely fouled with mineral deposits, the capillary pads were all removed and the filter house cleaned, sand-blasted, and painted with acid resistant paint. Several types of glass fiber filter media were then installed in various arrangements in the housing with continual air sampling and radioactivity tests being conducted. In the late fall of 1956, it appeared that a dry filter system consisting of a primary filter bank with two 1 in. layers of glass fiber media, and a secondary filter such as fine fiber glass media filters would reduce the counts out of the exhaust stack to a level acceptable to the Los Alamos Scientific Laboratory Health Division and the AEC Standards. At the present time, plans are underway and work is in progress to install the same system in additional wings as the operating needs dictate.

It should be pointed out that any system of exhaust ventilation can hardly be expected to perform efficiently and economically unless all of the factors concerning its operation are known at the time of design. In the case of the CMR Building, it was not possible during the design of the building to know accurately such things as acid fume and corrosive mist concentrations, particle size, density, and loading rate. After four years of operation in the building, these factors are now sufficiently well known so that a more accurate analysis of the problem can be made. Although such a solution to the problem is costly, this is one of the prices that must be paid for research work in an age as new and perplexing as the atomic energy era.

WARREN VIESSMAN, Baltimore, Md.: This paper on Radioactive Process Ventilation is very timely and advisedly brief, as it must of necessity be. I believe there are some 1500 research and industrial installations in this country, employing radioactive energy in which the principles outlined in this paper are applicable. That seems to be an astounding figure, as we are just entering this atomic age. I have no doubt however, that it is going to grow by leaps and bounds.

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‡ Los Alamos Scientific Laboratory.

In addition to what is evident and what the authors have said, there are two underlying principles in regard to purification. One is that the contaminated areas are under negative pressure so contaminated air does not leak out of the building or into other parts of the building. That is indicated by pressure figures on the diagram, but I believe if it is a design principle it should receive greater emphasis.

The other one is that in the figure that shows the connections, that is Fig. 3, the exhaust purification equipment is also under negative pressures. This equipment is made up of a number of filters for successfully purifying the atmosphere. I think it was intended that it be designed with a negative pressure so if there is a leakage between filter sections, the leakage would be into the filter rather than out of the filter, so all of the contaminated air must pass through the filters with the exhaust going out of the stack. If this is the case, if these are the two design principles, I would appreciate Mr. Glassmire commenting on them.

For the record, in regard to available purification equipment, there seems to be one item lacking. The capillary filters have been mentioned as a means of removing vapors and fumes and apparently have given excellent results, especially if the water is good or if there is water in sufficient quantity. I believe that activated charcoal or impregnated activated charcoal has also been in use in some installations. I would appreciate the author's comment on the inclusion of that medium for removal of gases and vapors from the radioactive processes.

The paper mentions protective shelters. I wish I knew how many protective shelters there are in this country. There are a great many, ranging from individual shelters that you might have in your own home, or adjacent to your home, to large shelters by industry and government. I know of one corporation in Pennsylvania that has 10 such shelters. Some are small while others are of considerable size. That seems like a large quantity for one company, but has doubtless been considered as a highly desirable means of protection for key personnel, equipment and processes.

The purification equipment that is used to purify the exhaust gases from a laboratory or radioactive process, is in general applicable to protective shelters. The process is however reversed. The purification equipment is applied to the air entering the shelter rather than to the air leaving the shelter, as is the case for the laboratory. Shelters are of several types. Some of them are completely inclosed with means for removing carbon dioxide, purifying the air and adding oxygen by one means or another. Shelters pressurized with ventilation outside air have absolute particulate filters and impregnated charcoal adsorbers for gases and vapors.

The shelters in this country afford varying degrees of protection against blast, temperature, chemical, biological and radiological attack. We may be a little behind in shelter construction in regard to blast and temperature protection, in comparison with some countries, as the necessity for development along these lines has not heretofore been as great here as elsewhere. In regard to protection against chemical, biological and radiological attack, our protective equipment is believed to be the best in the world.

J. H. CLARKE, Chicago, Ill.: From the previous comments, it is evident how important this paper is to industry at large. The authors have made a contribution which will be of use to a great many designers. Their efforts will be greatly appreciated. There are several points, however, which should be emphasized, because the problems of supplying make-up air to laboratories are not well understood.

One thing which I noticed in the paper was that the negative pressures maintained in the laboratories with respect to adjacent corridors were in the order of several hundredths of an inch. When it is realized that 100 feet per minute of air velocity corresponds to a velocity pressure of 0.0006 in. and that 2000 feet per minute is equivalent to 0.25 in. of velocity pressure, we can better understand why some of these construction specifications requiring from 0.25 in. to 0.50 in. of negative pressure in a laboratory are unrealistic. Even with double door air locks, such high velocities and pressure differentials are undesirable. The negative pressures maintained in the laboratories described in this paper are very realistic and probably have worked out well. Where portable type steel room

partitions are used, it is difficult, if not impossible, to maintain negative pressures in excess of 0.05 in. If the laboratory spaces are of tight construction, serious door trouble will result from high negative pressures.

It is also noted that the make-up air to the laboratories was introduced through perforated ceilings. In modern laboratories, the space air change may be in the order of a half minute to a minute. At that high supply rate, it becomes almost impossible to introduce the air into the space in any other way without disturbing the hoods and making them useless. The problem of getting the air into the laboratory spaces should always be studied with great care.

Another point made in the paper is that no attempt was made to design for static regain in the stacks discharging the effluent. There is nothing like a *lazy* stack to defeat the problem of getting rid of the effluent. Admittedly it is not the only, or the major, item involved in maintaining effective stack discharge, but high stack discharge velocity does help considerably to get rid of the effluent to the best possible degree.

W. W. HACKNEY, Dayton, Ohio: I picked up the reference to contaminated water used in spraying the ducts, so it would be of interest to know what is being done with it; also the reference to stainless steel for duct construction brings up the question,—what would have happened if the ducts had been made of ordinary inexpensive steel construction?

P. B. GORDON, New York, N. Y.: I would like to take this opportunity of complimenting the authors on their presentation. Certain questions come to mind regarding the exhaust. Would the author care to comment on the relative advantages of a central exhaust collecting system as compared with a modular exhaust collecting system, keeping in mind the problems of future flexibility, changing air patterns because of changing requirements within the laboratory and another important problem regarding system contamination, whereby an unusual occurrence in a specific laboratory might create sufficient contamination to require a shutdown of an entire laboratory wing in addition to having to clean up the entire exhaust duct?

The second general question has to do with the problem of exhaust static pressure control. Was the control set up on the basis of each controller working as a differential controller between one space and another space or was there a central basing point with all controllers acting to control differentials between the control point and the central controller or reference point? If so, in order to obtain a true reading of outdoor atmospheric pressure, where was this reference point located?

The third question concerns whether there is any specific advantage in locating some kind of rough cleaning filter at the hood to retain some of the contamination locally rather than let it pass out through the ducts, and assuming some of the contamination would drop out in the ducts before it reaches the central filters.

LEONARD D'OOGE, Sacramento, Calif.,: I have been hesitating to add anything to the many viewpoints and ideas we are giving our author because he already has a rather overwhelming assortment. Due to the fact that I have successfully handled some tough dust-particle problems with the electrostatic air-cleaner I wonder why it is not being used in this instance. They have used only the impingement and other mechanical types, if I heard correctly. I'm sure there is some good reason why—and I would like to know what it is.

AUTHORS' CLOSURE (Mr. Glassmire): I will not attempt to answer the questions or discuss the comments of the discussers in their order, but may skip around some since some points may have a bearing on others.

As regards the comments made by Mr. D'Ooge, I would say first of all that the reason why impingement type filters were used for final clean up in preference to other types at Los Alamos and at most Atomic Energy Commission installations, is that in the CMR Building the conditions were such that impingement type filters seemed to be called for. However, I think every kind of filter media is employed somewhere at the A. E. C. We have experimented with all types including settling chambers, cen-

trifugal separators, electrostatic precipitators and impactors and dry filters. We have most of these types in one place or another somewhere in the CMR Building.

As regards Mr. Gordon's discussion, I note that his first question concerns the advantages of a central exhaust over a modular or local exhaust and filtration. In my presentation I did not emphasize this, but in order to avoid undue contamination of the main filter towers, local filtration is also used. Some kind of a filtering system is put in right at the source. One type of hood used depends upon the particular matter being discharged and there is a little separate exhaust system and filtration setup right at the source if that source is an undue contributor to the total discharge.

With regard to another point in Mr. Gordon's discussion, I think it can safely be said that in this particular building the stress is on metallurgy. This is becoming more and more so. The particulate matter in the discharge stream is made up of heavy, irregular shaped, high specific gravity metallic particles. The dust and the contamination in the air stream are essentially of two types. Room dust has induced radioactivity on it as do also the radioactive particles themselves in the small submicron range and these particles are metallics.

Concerning the location of static pressure regulators, and the point Mr. Gordon made concerning the way pressure differentials are maintained in various chambers in reference to the static-pressure controls, the controls—the sensing elements—are in the mains. The static pressure is varied in both the exhaust mains and the inlet mains to give the desired overall static pressure condition. The static pressure differentials in each chamber are regulated by hand-feeding the exhaust until the differential is established at the desired point and making sure that there is a little more supply than exhaust. The static pressure differentials are not automatically controlled. With the number of rooms and laboratories involved, a static pressure controller can not be located in every place where a differential is wanted.

Mr. Engles made a welcome addition to the paper by emphasizing ultimate final efficiency of the clean up. Perhaps this was not especially emphasized in the paper. At the end of the exhaust a 99 percent plus efficiency is aimed at.

The remarks of Mr. Wherritt also came as a welcome addition to the paper. Perhaps in the near future a paper giving functional performance and operational data on this building may be prepared. The present paper aims to summarize the design of the system without going into too much of the performance data.

Mr. Hackney asked what is done with the contaminated water used in the wet cells and in the duct downwash. At Los Alamos there is a special treating plant which handles all liquid waste. The water used to wash the duct is tested and if it is above a certain tolerance level, it is sent to the special treating plant for treatment before being discharged on the land or back into the earth.

Mr. Hackney also asked what would happen if stainless steel had not been used. That is difficult to answer. Sufficient time has not elapsed to learn the life span of the ducts, but it was thought that by making them heavy and thick and of stainless steel that they would have a long life. As things have turned out, the acid condition is not as great as the building was designed for, however, with the high contaminated dust, long life for the duct should be a very important question, and if funds are available to make it of the best material possible, it is believed that thick steel will undoubtedly pay off in the end.

Mr. Viessman made the interesting comment that there are perhaps 1,500 other A.E.C. installations throughout the country and that there would be many more. That was one of the conclusions of this paper.

He also asked for comment on activated charcoal and similar substances as far as gaseous filtration is concerned. Both activated charcoal and similar media in various applications have been used and perhaps it has been used also as local filtration. There is also another process where use is being made of similar media for specific gas filtration and special chemical gases.

1604

No. 1604

SEMI-ANNUAL MEETING, 1957

MURRAY BAY, QUE., CANADA

With the Manoir Richelieu at Murray Bay, Que., Canada as the site, the 1957 Semi-Annual Meeting programmed a total of 6 sessions into three days. There were concurrent sessions each day, and committees met during the only otherwise unscheduled hours. The result was that the Meeting presented an exceedingly busy program. It also attracted an attendance of 412, made up of 222 members, 11 guests, 159 ladies, and 20 children.

As is the case at all Society Meetings, the success of the program (see pages 310 to 312) was due in great part to the preparatory work of the Committee on Arrangements and also to the care with which the planned arrangements were carried out.

The first session on Monday afternoon, June 24 at 2:00 p.m. was opened in the Rose Room of the Manoir Richelieu by Pres. P. B. Gordon, New York, N. Y., who called on D. Lorne Lindsay, chairman of the Committee on Arrangements who extended the greetings of the Montreal Chapter as host and who also outlined the several features of the program.

President Gordon also announced that the Committee on Resolutions would be made up of C. S. Koehler, New York, N. Y.; G. A. Linskie, Dallas, Tex.; and H. G. Gragg, Chicago, Ill.

The first programmed paper was presented by S. F. Gilman, Syracuse, N. Y., and following its discussion, President Gordon called attention to proposed amendments to the Society By-Laws. He had reference to the amendments printed June 24, 1957, as a result of their approval by the Council on June 23, 1957. The full statement of this draft of the proposed amendments is as follows:

ARTICLE II, ***Section 3. Qualifications.***

(*a*) *Honorary Member.* An Honorary Member shall be a notable person of preeminent professional distinction.

(*b*) *Presidential Member.* A Presidential Member shall be a Past President of the Society.

(*c*) *Life Member.* A Life Member shall be a Member, an Associate Member, or an Affiliate, in good standing, who has attained the age of sixty-five (65) years and has paid dues in said grades for thirty (30) years.

(*d*) *Fellow.* A Fellow shall be a MEMBER who has attained unusual distinction in the arts relating to the sciences of heating, ventilating, cooling or air conditioning, or

the allied arts and sciences, or in the teaching of major courses in said arts and sciences, or who by reason of invention, research, original work, or as an engineering executive on projects of unusual or important scope, has made substantial contribution to said arts and sciences, and who has attained the age of forty-five (45) years, and has been in good standing as a MEMBER for a period of at least ten (10) years prior to the date of his proposal for Fellow grade.

(*e*) *Member.* A Member shall be an engineer or teacher in engineering having ten (10) years or more of experience of a character satisfactory to the Council in the arts relating to the sciences of heating, ventilating, cooling or air conditioning, or the allied arts and sciences, of which four (4) years or more shall have been in responsible charge of important engineering work or in the teaching of courses in said arts and sciences; or an architect, chemist, physician, scientist or other person, deemed by the Council to be qualified by reason of special experience in the said arts and sciences.

Graduation from an engineering school with an engineering curriculum accredited by the Engineers' Council for Professional Development, or from an engineering school outside of the United States maintaining similar standards, or from an engineering school of collegiate grade approved by the Council shall be deemed equivalent to four (4) years of such experience; from other technical schools maintaining four (4) year courses, three (3) years; from a non-engineering college, two (2) years; and each successfully completed year in an engineering college shall be deemed equivalent to one (1) year of such experience.

(*f*) *Associate Member.* An Associate Member shall be a person having been graduated from a college or school of engineering with an engineering curriculum accredited by the Engineers' Council for Professional Development, or from an engineering school outside of the United States maintaining similar standards, or from an engineering school of collegiate grade approved by the Council, or possessing eight (8) years of experience satisfactory to the Council in the arts relating to the sciences of heating, ventilating, cooling or air conditioning, or the allied arts and sciences. Each successfully completed year in such college or school shall be deemed equivalent to one (1) year of such experience.

(*g*) *Affiliate.* An Affiliate shall be a person not possessing qualifications for Member or Associate member grades but whose pursuits, scientific attainments or practical experience qualify him to cooperate with heating, ventilating, cooling or air conditioning engineers in the advancement of engineering knowledge and practice.

(*h*) *Student.* A Student shall be a person registered as a graduate student or as an under-graduate student pursuing an engineering curriculum accredited by the Engineers' Council for Professional Development or in a college or school of collegiate grade approved by the Council. A student membership shall terminate upon the student's graduation, or at the close of the school term during which his enrollment as a student has ceased.

Section 4. ***Elections.*** Membership in the Society and advancement in membership grade shall be by vote of the Council on proposals or applications, or as set forth below. An unanimous vote by secret ballot shall be required for election to Honorary grade. Fifteen (15) votes by secret ballot shall be required for election to Fellow. A president of the Society shall advance to Presidential member upon installation of his successor. A two-thirds vote of the Council shall be required for election to other membership grades.

A person whose student membership has terminated may be advanced to Associate or Affiliate grade by the Council upon application providing proof of his qualifications therefor.

The grade of Honorary member shall be conferred on no more than three (3) persons in any calendar year. The grade of Fellow shall be conferred on no more than ten (10) MEMBERS in any calendar year. No member of the Council shall be elected to Honorary or Fellow grade.

Before election or advancement to Affiliate, Associate, or Member, the name of the applicant shall be published in an issue of the JOURNAL or mailed to all members.

Section 5. ***Proposals and Applications.*** Proposals and applications shall be on forms approved by the Council. Proposals for Honorary and Fellow grades shall be signed by no less than ten (10) MEMBERS, no more than five (5) of whom shall be members of the same Chapter of the Society. Members of the Council may not be proposers. No application shall be required for advancement to Life Member.

Section 6. ***Rights and Privileges.*** MEMBERS and Associate Members shall have the right to vote in the Society. MEMBERS shall be eligible to hold elective office in the Society.

ARTICLE VII, *Section 2.* ***Council Committees.***

Unless otherwise provided, the Council Committees and the respective chairmen thereof shall be appointed by the President, with the approval of the Council, as soon as practical after the close of the Annual Meeting. *Each Council Committee, except Regions Central Committee, shall consist of not more than five (5) members, of whom not less than three (3) shall be Council members. Members of the Committees shall serve for a term of one (1) year and until their successors are chosen.* The following shall be the Council Committees and their respective duties: (No change in sub-sections)

President Gordon explained that under the By-Laws, it was necessary to present these proposed amendments in order that they might be voted on at the next Annual Meeting of the Society. No additional amendments to the By-Laws were proposed.

The paper by G. O. Handegord, Saskatoon, Sask., Canada, was presented and discussed while a paper by J. A. Goff, Philadelphia, Pa., was presented by title only. President Gordon then adjourned the session.

At the second session held on Monday, June 24 at 2:00 p.m. in the Casino at the Manoir Richelieu, First Vice Pres. E. R. Queer, University Park, Pa., was the presiding officer.

This session had been organized as a Symposium on Air-Conditioning Instrumentation, with C. H. Pesterfield, East Lansing, Mich., as the symposium chairman. It had been expected that R. N. Pond, Rochester, N. Y., would act as moderator, but he found it impossible to attend. Professor Pesterfield, therefore, acted as moderator for the session.

The several speakers at this symposium session were members of the Instrument Society of America and that Society had cooperated with Professor Pesterfield in the planning and organization of the symposium. Professor Pesterfield acknowledged the gratitude of the Society at the excellent cooperation which had been received from the Instrument Society of America.

Four speakers had prepared papers for presentation as listed in the program.

The moderator called on each of these speakers successively and following their presentations, the session was thrown open for a discussion which was participated in by 5 of those in attendance.

Moderator Pesterfield announced that as usual with papers presented at symposium sessions, it is not planned to publish the papers or the discussions in the TRANSACTIONS. However, the full text of these papers can be found in the ASHAE JOURNAL SECTION, *Heating, Piping and Air Conditioning* as follows: the paper by V. Robert Baird on pps. 166–170 of the July 1957 issue; the papers by H. W. Alyea, L. F. Flagg and W. G. Young can be found on pps. 126–137 of the August 1957 issue.

At the conclusion of this announcement, 1st Vice Pres. Queer closed the session at 4:30 p.m.

The third session held on Tuesday morning, June 25 at 9:30 a.m. had been organized as a topical session on Sound and Vibration. Following the opening of the session by 2nd Vice Pres. A. J. Hess, Los Angeles, Calif., he introduced H. A. Lockhart, Morton Grove, Ill., to act as chairman during the presentation of the papers.

The four papers for this session were then presented and discussed, following which Second Vice President Hess adjourned the session.

The fourth session met in the Casino on Tuesday, June 25 at 9:30 a.m. and was called to order by Treas. C. H. Pesterfield, East Lansing, Mich. Following the presentation and discussion of the four papers which had been programmed for this session, it was adjourned.

The fifth session on Wednesday, June 26 at 9:30 a.m. was held in the Casino at the Manoir Richelieu and was opened by 1st Vice Pres. E. R. Queer. The session had been organized in the form of a Symposium on Sound and Vibration with John Everetts, Jr. as symposium chairman. J. B. Graham, Buffalo, N. Y., was present to act as moderator for the session for which 5 papers as listed in the program had been prepared.

In accordance with the usual procedure at symposium sessions, there was a discussion period following the presentation of the last paper.

Moderator Graham announced that there are no plans for publishing these 5 papers in the TRANSACTIONS. The full text of each paper is published in the ASHAE JOURNAL SECTION, *Heating, Piping and Air Conditioning* as follows: the paper by R. J. Wells on pps. 138–143 of the August 1957 issue; the papers by R. N. Hamme, H. C. Hardy, F. B. Holgate and S. Baken, and R. E. Parker can be found on pps. 143 to 166 of the September 1957 issue. He also announced that consideration is being given to the collection of the several papers into a special bulletin.

Following the close of the discussion period, President Gordon took charge of the session and read the Report of the Nominating Committee. He then called on C. S. Koehler, New York, N. Y., chairman of the Resolutions Committee who read the Report of that Committee which is printed here in full.

RESOLUTIONS

WHEREAS, the Semi-Annual Meeting 1957 of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS is now drawing to its close, and

WHEREAS, this meeting was held at Murray Bay, Quebec, Canada, in the land of the historical past, expanding present and radiant future, where the American dollar felt the inflated "holler" of the 5 percent Canadian touch, and

WHEREAS, the technical life of the Society has been advanced through the presentation and discussion of a series of well-prepared papers and conflicting scheduled symposia, and

WHEREAS, through the superb facilities provided, the technical and other committees of the Society have been able to conduct their meetings and accomplish their aims efficiently in accommodations well air-conditioned—by nature, and

WHEREAS, the Montreal Chapter, its members and Committee on Arrangements have been especially generous hosts to members and guests by serving tea and cocktails, *largesse de coeur*, and

WHEREAS, the host chapter and guests have been exceptionally gracious to our Washington inaugurated member, Mabel Henderson, therefore:

BE IT RESOLVED that we who have had the privilege to attend this pleasant meeting extend our sincere appreciation and gratitude:

TO the Montreal Chapter, its Officers, Board of Governors, and members for their generous support,

TO D. Lorne Lindsay, general chairman, and the vice chairmen, and all other members of the Committee on Arrangements,

TO Mrs. A. E. Horsburgh and the ladies of the Chapter for their gracious hospitality in entertaining the visiting ladies and families,

TO Mr. J. P. Fitzsimons for special children's events and activities,

TO the authors and discussers of the technical papers and symposia for their educational contributions,

TO Mr. F. Aylmer Hamlet who so capably served as Toastmaster of the banquet,

TO Dr. David L. Thomson, Dean of Faculty of Graduate Studies and Vice-President at McGill University for his informative address on the Dominion of Canada,

TO Manoir Richelieu management and its staff for outstanding courteous and efficient service,

TO members of the headquarters staff under the direction of A. V. Hutchinson whose untiring effort is always appreciated,

TO President P. B. Gordon whose wisdom and energy has expanded the scope and stature of the Society,

TO the other Officers, members of Council and committees of the Society for their excellent contribution to the aims and objectives of the Society in this our 63rd year.

Respectfully submitted,

The Resolutions Committee

C. S. KOEHLER, *Chairman*, New York, N. Y

G. A. LINSKIE, Dallas, Tex.

H. G. GRAGG, Chicago, Ill.

Following adoption of the Report, and there being no unfinished or new business to come before the meeting, President Gordon adjourned the Meeting.

The sixth session which was held on Wednesday, June 26 at 9:30 a.m. took place in the Rose Room at the Manoir Richelieu and was opened by Pres. P. B. Gordon. He successively called for the presentation of the 3 papers which had been programmed for the session.

Following the presentation and discussion of these 3 papers, President Gordon announced that the remaining items on the program consisted of the Report of the Resolutions Committee, any unfinished business and any new business and these would be taken up in the fifth session in the Casino. He urged that all interested go to the Casino to take part in this part of the meeting. President Gordon then closed the sixth session.

PROGRAM—SEMI-ANNUAL MEETING

Manoir Richelieu, Murray Bay, Canada

June 24-26, 1957

Saturday—June 22

1:30 p.m. Executive Committee (*Card Room* 1) E. R. Queer, *Chairman*
3:30 p.m. Regions Central Committee (*Card Room* 2) A. J. Hess, *Chairman*
9:30 p.m. Dancing (*Casino*)

Sunday—June 23

9:30 a.m. Finance Committee (*Card Room*) Walter A. Grant, *Chairman*
10:00 a.m. REGISTRATION (*East Lounge*)
1:30 p.m. Council Meeting (*Murray Room*)
2:00 p.m. Research Executive Committee (*Card Room*) H. A. Lockhart, *Chairman*
3:30 p.m. Welcome Tea
8:00 p.m. TAC on Insulation (*Murray Room*) M. W. Keyes, *Chairman*
8:00 p.m. TAC on Plant and Animal Husbandry (*Card Room*) A. J. Hess, *Chairman*
9:15 p.m. Movie (*Casino*)

Monday—June 24

9:30 a.m. REGISTRATION (*East Lounge*)
9:00 a.m. TAC on Air Cleaning (*Charlevoix Room*) E. F. Snyder, Jr., *Chairman*
9:00 a.m. TAC on Evaporative Cooling (*Murray Room*) Leo Hungerford, *Chairman*
9:00 a.m. TAC on Sorption (*Conference Room*) G. L. Simpson, *Chairman*
9:30 a.m. Program and Papers Committee (*Card Room* 1) John Everetts, Jr., *Chairman*
10:00 a.m. Building Committee (*Card Room* 2) A. J. Hess, *Chairman*
10:00 a.m. Ladies Putting Contest
2:00 p.m. FIRST SESSION (*Rose Room*)
Call to Order—Pres. P. B. Gordon

Greetings by D. Lorne Lindsay, *Chairman*, Committee on Arrangements

Thermal Circuit Analysis for Developing Application Engineering Information, by S. F. Gilman and O. W. Clausen, Syracuse, New York, presented by Mr. Gilman

Thermal Performance of Frame Walls, Part III—Wall Surface Temperatures, by G. O. Handegord, Saskatoon, Sask., Canada, presented by Mr. Handegord

Saturation Pressure of Water on the New Kelvin Temperature Scale, by J. A. Goff, Philadelphia, Pa. (by title only)

2:00 p.m. SECOND SESSION (*Casino*)
Call to Order—1st Vice Pres. E. R. Queer

SYMPOSIUM ON AIR-CONDITIONING INSTRUMENTATION
C. H. Pesterfield, Symposium Chairman
R. N. Pond, Moderator

Fundamentals of Control—V. Robert Baird
Control of High-Velocity Single-Duct Systems—W. G. Young
Control of High-Velocity Double-Duct Systems—H. W. Alyea
Coordination of High-Velocity Design—L. F. Flagg

2:30 p.m. Ladies Fashion Show and Tea

7:00 p.m. Buffet Supper (*Main Dining Room*),
Habitant Evening with Dancing (*Casino*)

8:00 p.m. Nominating Committee (*Card Room*)

Tuesday—June 25

9:00 a.m. REGISTRATION (*East Lounge*)

9:30 a.m. THIRD SESSION (*Rose Room*)
Call to Order—2nd Vice Pres. A. J. Hess

TOPICAL SESSION ON SOUND AND VIBRATION
H. A. LOCKHART, Chairman

Criteria for Room Noise from Air Conditioning—C. M. Ashley, Syracuse, New York, presented by Mr. Ashley

An Evaluation of Four Methods for Determining the Sound-Power Output of a Fan—W. F. Kerka, Cleveland, Ohio, presented by Mr. Kerka

Acoustic Power Level Determination for Machinery in Hard Rooms—W. J. Burch, Annapolis, Md., presented by Mr. Burch

Studies in Silencing a Diesel Driven Heat Pump—H. H. Vickers and R. W. Sage, Linden, N. J., presented by Mr. Sage

9:30 a.m. FOURTH SESSION (*Casino*)
Call to Order—Treas. C. H. Pesterfield

Acoustic Coupling of Residential Furnaces with Their Surroundings, by C. F. Speich, W. R. Dennis and A. A. Putnam, Columbus, Ohio, presented by Mr. Speich

Performances of Intermittently-Fired Oil Furnaces, by W. G. Colborne, Kingston, Ont., Canada, presented by Mr. Colborne

Measurement of Infiltration in Two Residences, Parts I and II, by D. R. Bahnfleth, T. D. Moseley and W. S. Harris, Urbana, Ill., presented by C. M. Humphreys, Cleveland, Ohio

Design and Performance of a Portable Infiltration Meter, by C. W. Coblentz and P. R. Achenbach, Washington, D. C., presented by Mr. Coblentz

9:30 a.m. Ladies Sightseeing Tour

1:30 p.m. Golf Tournament

2:30 p.m. Ladies Handicraft Exhibition Demonstration

7:00 p.m. SEMI ANNUAL BANQUET (*Main Dining Room*)

Toastmaster: F. Aylmer Hamlet

Speaker: David L. Thomson, Dean of Faculty of Graduate Studies and Vice-Principal, McGill University

Subject: Welcome To Canada

Presentation of Past President's Memory Book to John W. James by President P. B. Gordon

Presentation of Golf Trophies:
Past Presidents Research Bowl
Eichberg Memorial Cup
Paul Bunyan Cup

10:00 p.m. Dancing (*Casino*)

Wednesday—June 26

9:30 a.m. FIFTH SESSION (*Casino*)
Call to Order—1st Vice Pres. E. R. Queer

SYMPOSIUM ON SOUND AND VIBRATION
John Everetts, Jr., Symposium Chairman
J. B. Graham, Moderator

Noise Control Problems in Air-Conditioning Equipment—R. J. Wells

Techniques of Sound-Power-Level of Unitary Equipment—R. N. Hamme

Sound Standards for Testing and Rating—H. C. Hardy

Application of Standards to Equipment—R. E. Parker

A Method for Estimating Octave Band Level of Noise Generated by Air-Conditioning Systems—F. B. Holgate and Sidney Baken

9:30 a.m. SIXTH SESSION (*Rose Room*)
Call to Order—Pres. P. B. Gordon

Performance Testing of Roof Ventilators, by J. J. Wannenburg, Pretoria, South Africa, presented by R. G. Huebscher, Cleveland, Ohio

Jet Velocities from Radial Flow Outlets, by Alfred Koestel, Cleveland, Ohio, presented by Mr. Koestel

Velocities and Temperatures on Axis of Downward Heated Jet from 4-Inch Long-Radius *ASME* Nozzle, by Rudiger Knaak, Bremen, Germany, presented by Linn Helander, Manhattan, Kans.

Report of Resolutions Committee
Unfinished Business
New Business
Adjournment

10:00 a.m. Ladies Handicraft Exhibition Demonstration

2:00 p.m. TAC on Air Distribution (*Murray Room*)
W. O. Huebner, *Chairman*

COMMITTEE ON ARRANGEMENTS

D. LORNE LINDSAY, *General Chairman*

Vice Chairmen

W. G. HOLE, B. J. HORSBURGH, R. J. KER, H. G. S. MURRAY

Honorary Chairmen

F. AYLMER HAMLET G. LORNE WIGGS

Reception and Registration: G. Lorne Wiggs (*Honorary*), and C. D. Lynch
Ladies: Mrs. A. E. Horsburgh
Children: J. P. Fitzsimons
Transportation and Hotel Reservations: W. E. Jarvis
Finance: F. G. Phipps (*Honorary*), Ralph Grossman
Banquet: Leo Garneau
Entertainment: G. N. Martin, W. G. McCrudden
Golf: W. J. McAdam
Publicity: W. U. Robinson
Sessions: Eric Williams, Marcel P. F. Turnau

1605

No. 1605

THERMAL CIRCUIT ANALYSIS FOR DEVELOPING APPLICATION ENGINEERING INFORMATION

By STANLEY F. GILMAN* AND O. WILLIAM CLAUSEN**, SYRACUSE, N. Y.

IN DESIGNING the air-conditioning system for a commercial building, the objective of the application engineer is to select refrigeration equipment that will closely match the peak cooling load. However, accurate evaluation of this peak load is difficult. The outdoor temperature and the position of the sun in the sky are continually changing. Lights are turned on at the beginning of the work day and off at the end. Radiation portions of heat gains are stored in the structure and released later. Hence the problem is one of unsteady-state heat flow, which even under simple conditions can be very difficult to solve.

Temporary storage of heat in structures and furnishings complicates evaluation of the actual equipment (cooling) load. Consequently, heat gains are generally considered to be cooling loads; *e.g.*, the entire sun heat gain and heat equivalent of the lighting wattage are presumed to appear immediately at the refrigeration equipment. The effect of thermal storage is to reduce the refrigeration capacity required for a particular installation; when radiation heat gains are large the reduction is considerable. Therefore, it is essential that application engineers have data properly accounting for thermal storage effects.

During the past few years, the problem of thermal storage has been receiving more attention, undoubtedly stimulated by the 1949 paper of Mackey and Gay[1]. Since that time several ASHAE papers have been written. Further impetus was provided by 2 papers by Nottage and Parmelee of the ASHAE Research Laboratory[2,3]. They promoted the concept of representing the thermal problem by a *thermal circuit diagram* using electrical symbols for the equivalent thermal properties. Of particular significance, it was shown that the resultant network could be solved without recourse to higher mathematics. In addition, many problems could be solved by manual computation; *i.e.*, large analogue or digital computers were not essential for solving the equivalent electrical networks.

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* Senior Research Engineer, Research Department, Carrier Corporation. Member of ASHAE.
** Research Engineer, Research Department, Carrier Corporation.
[1] Exponent numerals refer to References.
Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Murray Bay, Que., Canada, June 1957.

The most recent ASHAE laboratory paper[4] presents an analysis of a typical commercial office building. From an application engineer's viewpoint, this circuit would certainly be regarded as complex. Therefore, a basic problem is to devise means by which application engineers can profit from the thermal circuit method of cooling load analysis.

The key to exploiting thermal circuit analysis seems to be in training a small nucleus of engineers in the method. These engineers could be given the assignment of developing application engineering information of a general nature. Ideally, this would consist of simple multiplying factors to apply to the present

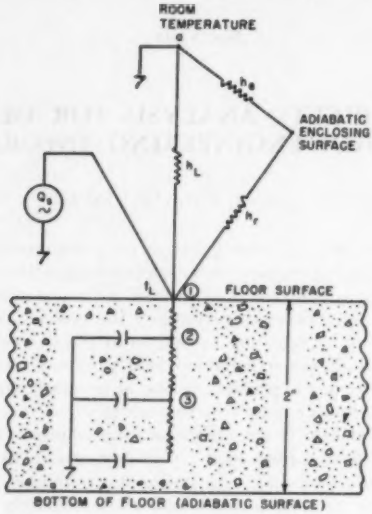

FIG. 1—THERMAL CIRCUIT DIAGRAM OF 2-IN. CONCRETE FLOOR PROBLEM

load calculation data. In any case, the information must be presented in a simple form that can be used quickly to select equipment for any type or size of building, no matter what its orientation or geographical location.

A second responsibility of the group would be to make thermal circuit analyses of special field problems that are ordinarily sent into the home office.

The purpose of this paper is to report the experience of the authors' company in learning and utilizing the thermal circuit method of analysis. As would be expected, the experience has disclosed merits and limitations of both the method and the particular procedure used to solve the circuits. These are reported in some detail. In addition, procedures for developing extensive application engineering information are described, as well as some of the difficulties yet to be overcome.

SIMPLER PROBLEMS

To gain familiarity with the mechanics of the procedure of thermal circuit analysis, initial problems were simple ones having known solutions. A 4-in.

gypsum panel problem which appeared in a 1932 ASHAE paper[5] was solved with the panel divided into 2, 3 and 4 lumps. Results were compared with Fig. 31 of the paper[5]. Agreement was excellent with 4 lumps, good with 3 lumps, and only fair with 2 lumps. Although it was known that the accuracy improves as the number of lumps is increased, experience was gained in predicting the *influence* of the number of lumps on the solution.

The procedure given in Table 5 of this work[5] for expressing the temperature input as a Fourier series was followed. Calculations and comparisons showed that 24 hourly temperatures were sufficient, rather than the 72 used by Houghten[6],

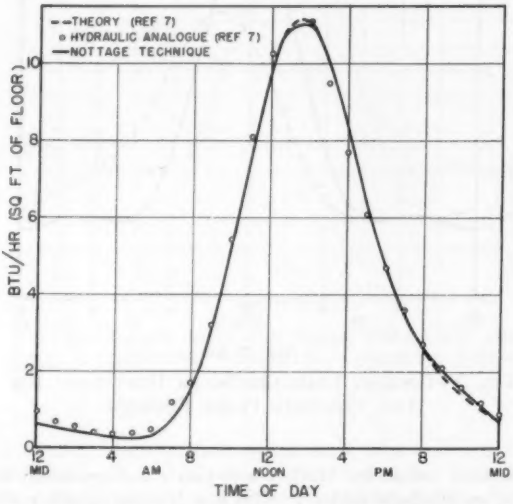

FIG. 2—COOLING LOADS FOR 2-IN. CONCRETE FLOOR PROBLEM AS OBTAINED BY THREE METHODS

thus reducing the subsequent calculations to one-third. The number of harmonics needed for good representation is also important, since each harmonic requires a separate solution of the network. For symmetrical waves, such as outside surface temperatures of flat roofs, 2 or 3 harmonics are usually sufficient. For non-symmetrical waves; *e.g.*, east and west wall outside surface temperatures, 3 to 5 are often needed.

The procedure described by Nottage and Parmelee[2,3] was followed throughout. Each point in a network at which a thermal current balance is made yields 2 linear algebraic equations. Hence a 9-point circuit yields 18 linear equations in 18 unknowns to be solved simultaneously. In this paper, the technique used by Nottage is used to denote their method[2,3] of solving the network, as contrasted with solving it by analogues or other devices or procedures.

By restricting the initial circuits to a maximum of 5 junction points, hence 10 equations, solutions were obtained for several other sample problems by manual

computation with desk-type calculators. Sets of 10 equations or less are readily solved by available arithmetical techniques[6]. Results were checked by substitution back into the original equations.

After considerable experience with single walls and roofs, a more involved circuit was set up which included radiation exchange between surfaces. The prob-

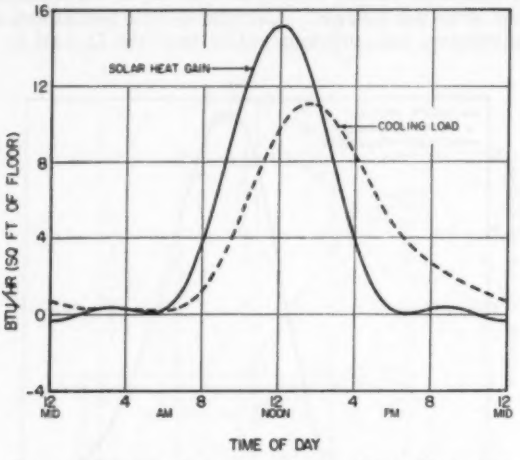

FIG. 3—COOLING LOAD AND SOLAR HEAT GAIN FOR 2-IN. CONCRETE FLOOR PROBLEM

lem was one already solved by Mackey and Gay[7], and presented the additional complication of an adiabatic surface. This is a boundary surface through which heat does not flow in either direction, and requires consideration of a resistance-capacitance series branch. Selection of this problem also provided opportunity to compare the solution from the technique used by Nottage with both the theoretical solution and results from an hydraulic analogue[7].

The problem involves solar radiation falling uniformly on a concrete floor, and a surrounding adiabatic enclosure. The thermal circuit diagram is presented in Fig. 1. The floor has been divided into 3 lumps of the same thickness. Heat is removed at point *a* at just the rate to maintain the room temperature constant.

Current balances were made around points 1, 2 and 3 in Fig. 1, and the circuit equations were solved. Results are presented in Fig. 2. Except where 2 lines appear, the *theory* and curves from the technique used by Nottage are one. The agreement is therefore excellent. Also, the hydraulic analogue results[7] compare favorably with the curves.

Next, the floor thickness was increased to 6 in. and the thermal circuit solved again. Comparison of the cooling load curve with that of Mackey[7] again showed excellent agreement.

To illustrate the practical aspects of the results, Fig. 3 shows both cooling load and heat gain curves for the 2-in. concrete floor. The peak cooling (equipment)

load is only 73 percent of the peak heat gain and occurs 1½ hours later. By standard calculation procedures, 12 noon would be taken as the time of the peak load. However, at noon the cooling load is only 64 percent of the heat gain.

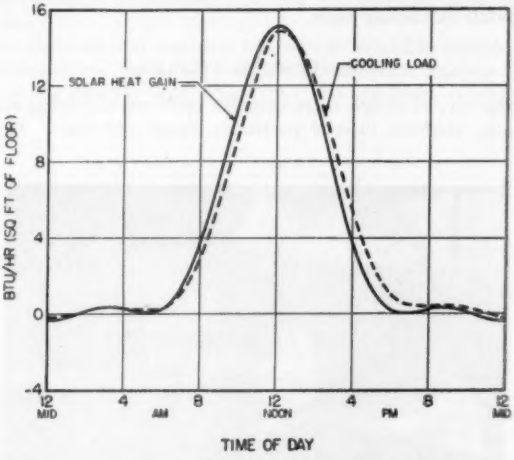

FIG. 4—COOLING LOAD AND SOLAR HEAT GAIN FOR 2-IN. VERMICULITE AGGREGATE CONCRETE FLOOR PROBLEM

For the 6-in. thick concrete floor, comparisons were even more striking. At noon the cooling load was only 55 percent of the heat gain.

Results of the thermal circuit analysis of the Mackey problem[7] brought conviction of the validity of the technique used by Nottage and that it was being properly applied to periodic heat flow problems.

EXTENSION OF RESULTS

In concrete construction, vermiculite is sometimes used as aggregate in place of stone or gravel. The resultant mixture is lightweight and has high thermal resistance, whereas conventional concrete is heavy and has low thermal resistance. To study the effect of vermiculite, it was only necessary to re-solve the circuit after assigning new values to the resistors and capacitors buried in the floor surface (see Fig. 1). Results for the 2-in. floor are presented in Fig. 4 and are striking in that, for engineering purposes, there is no thermal storage. The peak cooling load is 98 percent of the peak heat gain and the time lag is negligible.

When the 6-in. floor was similarly treated, the peak cooling load was found to be 87 percent of the peak heat gain. Thus, the conventional concrete floor 2 in. thick had more thermal storage than a 6-in. floor with vermiculite aggregate.

Valuable information was obtained from these thermal circuit analyses. Thermal storage in 2-in. concrete floors with vermiculite aggregate was found to be

negligible. For engineering purposes, the solar heat gain and cooling load are the same for thicknesses less than about 3 in. Thermal storage in conventional concrete construction is considerable; hence, application engineering data should clearly differentiate between these 2 constructions. Therefore, this additional experience with thermal circuit analysis indicated the potential influence of construction materials on cooling loads.

MORE COMPLEX PROBLEMS

The next step was to attack more complex problems involving spaces enclosed by outside walls, windows, interior partitions, floors and roofs. For such spaces

FIG. 5—650 MAGNETIC DRUM, DATA-PROCESSING MACHINE

it was soon found that the resultant thermal circuits involved a minimum of 8 junctions, and that usually more were desirable. Therefore, utilization of the technique used by Nottage posed the problem of finding economical means of solving 16 or more equations simultaneously.

For some sets of 28 equations, a special method was devised based on mathematical *relaxation* principles[8]. Although this method was handled easily by computation personnel with only desk-type calculators, it required about one week's work per harmonic. Therefore, for many of the problems 3 weeks to a month elapsed before the solution became available. Since the next step in the analysis often depended on the solution of a previous circuit, the time required to solve circuits slowed progress on the program.

Fortunately, at about this time a commercial 650 computer was delivered to the company. This is a magnetic drum type, card programmed digital computer (Fig. 5). The person seated is at the main console which contains the control panel (visible), magnetic drum, and electronic circuits for performing the arithme-

tic operations. The drum can store 2000 ten-digit numbers in its *memory* for an indefinite time. To the rear of the console is the power supply unit. In the right foreground is the input-output unit in which instructions and data punched on cards are placed. Problem solutions come out of this unit as punched cards, which are then placed in an auxiliary machine that converts holes in the cards to printed numbers.

Investigation disclosed that a general routine was available whereby any number of simultaneous equations, up to 41, could be solved by the machine. The time to solve a set of 28 equations is slightly less than one-half hour, or roughly *1 percent*

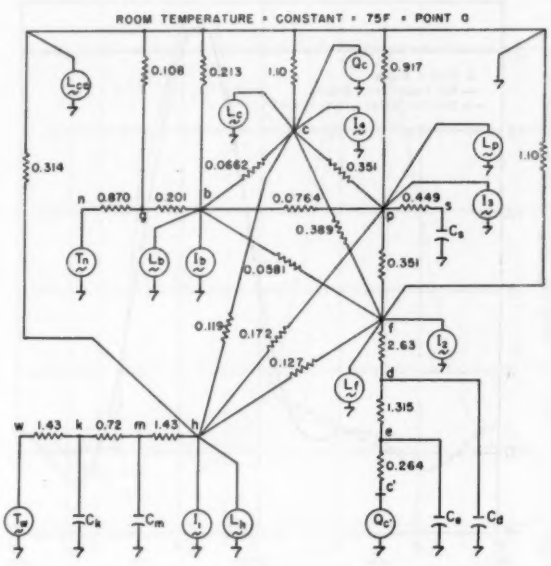

FIG. 6—THERMAL CIRCUIT DIAGRAM OF OFFICE PROBLEM

of the time by manual computation using relaxation methods. The machine made possible the rapid solution of several thermal circuits that were subsequently developed.

The most recently reported work has been application of the technique used by Nottage to a typical office for which the solution was obtained by a differential analyzer[4]. This particular problem was selected to acquire experience with thermal circuits involving blinds, plus the added complexity of a cooled ceiling panel. In addition, an opportunity was provided to compare the solution by the technique used by Nottage with that of the differential analyzer, so as to better evaluate the potential of analogue computers for solving cooling load problems.

In the paper[4], validation of the analyzer solution is based upon the agreement between calculated and analyzer steady-state values. Apparently either time or

a machine was not available for solving the sets of 18 equations for each harmonic. The set of equations for the steady-state solution could be solved by manual computation with little difficulty. In any case, agreement in steady-state values does not represent a complete check. Although the steady-state solution fixes the area under the cooling load *vs.* time curves, the *shapes* are not determined. Also, capacitors are not involved in the steady-state solution so the accuracy of the analyzer in duplicating the given capacitance values is not known. Hence, an additional objective was to complete the technique used by Nottage solution of this problem.

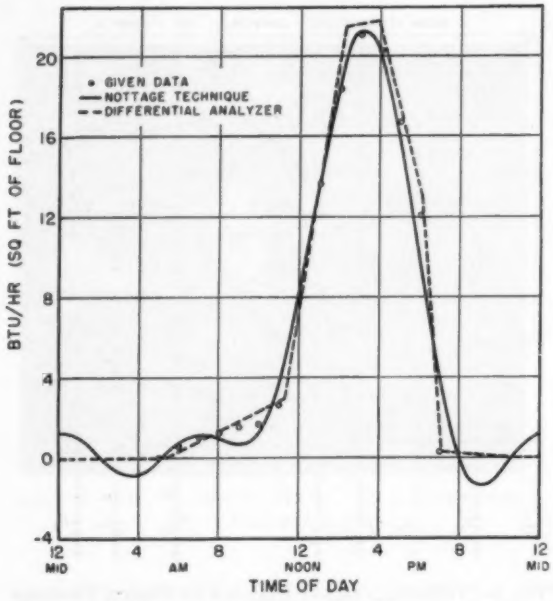

FIG. 7—COMPARATIVE REPRESENTATIONS OF SOLAR RADIATION IMPRESSED ON VENETIAN BLINDS

Complete details of the office are given in Reference 4. The 12 x 25 ft room with an 8 ft ceiling faces southwest. The gross exposed wall area is 200 sq ft of which 135 is wall and 65 is window equipped with a typical venetian blind. Typical solar radiation and outdoor temperature cycles were impressed on the office, as well as a lighting load of 3.6 watts per sq ft between 8 a.m. and 8 p.m.

The equivalent electrical circuit of the thermal problem is shown in Fig. 6. The lettering system for circuit junction points of Fig. 1, Reference 4, has been maintained. Numerical values of admittance per sq ft of floor area are shown at resistors. Letter symbols are used for capacitors since their values are frequency dependent. Current balances are made at 9 junction points in the network; con-

sequently, the technique used by Nottage yields sets of 18 simultaneous equations to be solved.

After drawing the circuit diagram, the given solar radiation input data[4] were expressed as a Fourier series. The steady-state value plus the first to third harmonics were calculated. Results are shown by the solid line in Fig. 7 for comparison with the given data represented by open circles. Agreement is seen to be excellent between 4 a.m. and 8 p.m.

The straight-line segments forming the dashed curve are those generated electronically for inputs to the differential analyzer. The fit with the given data is also very good.

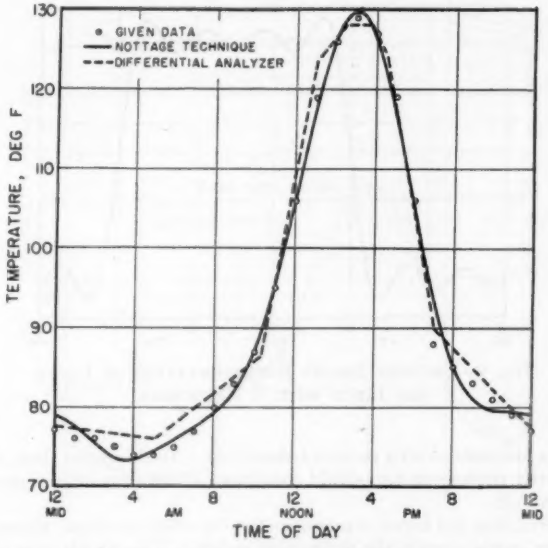

FIG. 8—COMPARATIVE REPRESENTATIONS OF SOL-AIR TEMPERATURES IMPRESSED ON EXPOSED WALL

It should be stated that the data for all differential analyzer curves were obtained directly from the ASHAE Research Laboratory.

Comparative input data for the sol-air temperature cycle on the outside wall are presented in Fig. 8. The Fourier series representation required for the technique used by Nottage fits the data very well and somewhat better than the analyzer.

Fitting data by linear segments, as done for this analyzer, is not the only way of representing input data for analogue computers. Since a Fourier series is basically the sum of sine waves of related frequencies, electronic oscillators provide another possibility. The analogue curve could then be identical to the curve representing the technique used by Nottage in Fig. 8. In addition, electronic function genera-

tors are available which will fit a continuous curve to data. In general, then, it is concluded that analogue computer inputs can accurately simulate given data.

The radiation portion of the impressed lighting load is shown by the dashed line of Fig. 9. When the lights are turned on at 8 a.m. the value jumps from zero to above 12; at 8 p.m. it jumps back to zero. Mathematically these jumps are finite discontinuities in the curve. Although such curves can be represented by Fourier series, for the same number of harmonics the fit is ordinarily not as good as with continuous functions. The solid line shows the Fourier series representation when the first 4, odd, harmonics are used. The fit is satisfactory.

As part of a research program, our organization has collected considerable test data from office buildings. These data show that the lighting wattage increases

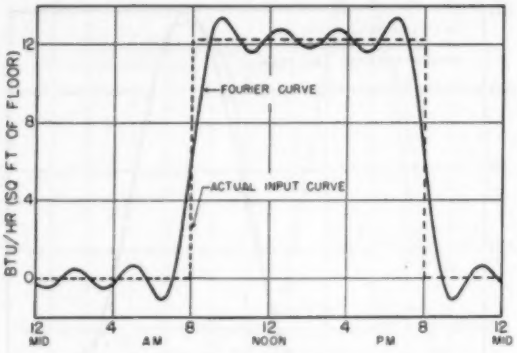

FIG. 9—FOURIER SERIES REPRESENTATION OF LIGHTING INPUT WITH 4 HARMONICS

from zero to a maximum over a period of about 1 hr. In this sense, then, the Fourier curve is a better representation of field conditions at the time office lights are being turned on and off.

After determining the input expressions for the office problem, thermal current balances were made around the 9 junction points. The circuit was solved with only the lighting load input, and then with all other inputs combined. The 18 equations resulting from the lighting input are shown in matrix form in Table 1. Within the matrix, letter symbols represent frequency dependent coefficients; their values are listed in Table 2. Solution of these by the 650 computer yielded values for the unknowns a_k, b_k, and so on, listed along the top of Table 1.

Temperature and heat flow curves corresponding to those in Reference 4 were computed and analyzed. The panel and air system cooling loads resulting from all inputs are shown in Fig. 10. Comparison of the technique used by Nottage and differential analyzer results show them to be in good agreement. The area under the dashed-line curve is slightly less than under the solid line curve. This means the 24-hr average (steady-state) value for the analyzer is less, which agrees with Table 2 of Reference 4.

Small differences in steady-state values are not serious. The important factor is the relative *shapes* of the load *vs.* time curves, and reference to Fig. 10 shows re-

markable agreement. Both curves have maximum values at 4 p.m. The sharp peak in the analyzer curve is caused by the linear segment inputs. The shape of each temperature and heat flow curve for individual surfaces and inputs, as obtained from the solution obtained from the technique used by Nottage, was very similar to the individual analyzer curves.

From this work it was concluded that analogue computers such as that described in Reference 4 have the potential of providing accurate solutions of complex cool-

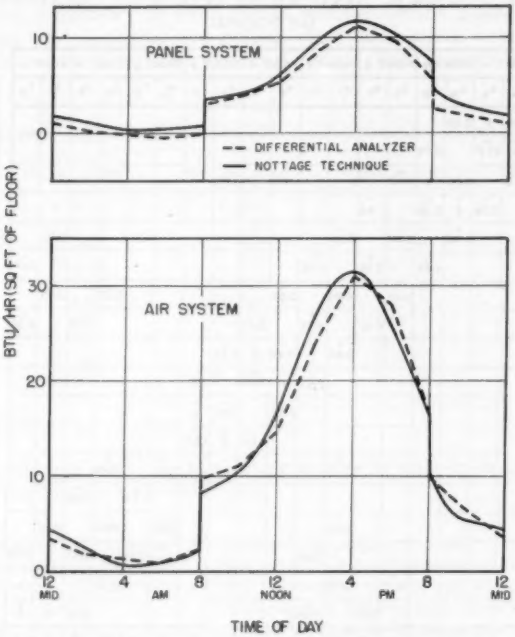

FIG. 10—COMPARATIVE AIR AND PANEL SYSTEM LOADS FOR OFFICE PROBLEM

ing load problems. At present, several different analogue computers are being studied to determine their capabilities for handling diversified types of heating and air-conditioning research problems.

MERITS AND LIMITATIONS OF THE TECHNIQUE USED BY NOTTAGE

Experience with the thermal circuits discussed, as well as many additional ones, has disclosed several merits of thermal circuit analyses and of the technique used by Nottage in particular. Although the procedure in the original paper[2] appears formidable, it has been found that engineers become adept in circuit analysis

after working a few problems. Circuit diagrams such as Fig. 6 are very helpful in visualizing the thermal problem. They also aid in predicting the influence of particular factors; for example, a high admittance value indicates an easy heat flow path.

One limitation of the technique used by Nottage is the amount of labor involved in solving several sets of equations simultaneously when only desk-type calculators are available. This limitation was overcome by the 650 computer. In

TABLE 1—TYPICAL ARRANGEMENT OF CIRCUIT EQUATIONS TO BE SOLVED

	POINT k		POINT m		POINT h		POINT f		POINT d		POINT e		POINT g		POINT b		POINT p		
	a_k	b_k	a_m	b_m	a_h	b_h	a_f	b_f	a_d	b_d	a_e	b_e	a_g	b_g	a_b	b_b	a_p	b_p	Q
1	−2.18	−A	+0.72																0
2	−A	+2.18		−0.72															0
3	+0.72		−2.18	−B	+1.43														0
4		−0.72	−B	+2.18		−1.43													0
5			+1.43		−2.18		+0.12										+0.17		0
6				+1.43		+2.18		−0.12										−0.17	−H
7					−0.12		−4.68		+2.63						−0.05		+0.35		0
8						−0.12		+4.68		−2.63						−0.05		−0.35	−K
9							+2.63		−3.94	−C	+1.31								0
10								−2.63	−C	+3.94		−1.31							0
11									+1.31		−1.57	−E							0
12										−1.31	−E	+1.57							0
13													−1.17		+0.20				0
14													+1.17			−0.20			0
15							−0.05						−0.20		−0.81		+0.07		0
16								−0.05						−0.20		+0.81		−0.07	−L
17					+0.17		+0.35								+0.07		−F	−G	0
18						−0.17		−0.35								−0.07	−G	+F	−M
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	

addition, the computer eliminated a considerable amount of labor during other phases of a thermal circuit analysis. The machine was programmed to develop Fourier series expressions when given 24 hourly values of temperature or heat flow. With the given input data, the machine calculated the Fourier coefficients for any or all of the first 10 harmonics. For the usual input function, machine time was about 1 minute.

With another 650 computer program, temperatures or heat flows as a function of time were computed using the previously determined Fourier coefficients. Plotting these and the input data on the same graph permitted the closeness of the Fourier series fit to be appraised. This same program was used to take the results from the simultaneous equation solution and compute the final temperatures and heat flows. The time required was also 1 min.

By utilizing the 650 computer, engineering effort was primarily devoted only to diagramming the thermal circuit, formulating the equations (see Table 1), and analyzing the results. Indeed, after some experience, computation personnel became adept at making the current balances and writing the equations.

With the multiplicity of details involved in a thermal circuit analysis, the probability of making errors is high. Therefore, means of avoiding errors were studied. The solution of each set of simultaneous linear equations was checked by substituting the answers back into the original equations. Circuit constants; *i.e.*, admittance values, could be checked individually.

The greatest probability of error was in making current balances, expanding and separating the equations, and getting them into matrix form. Study disclosed that the matrix of the coefficients had distinct characteristics that facilitated

TABLE 2—NUMERICAL VALUES FOR FREQUENCY DEPENDENT ELEMENTS IN TABLE 1

ELEMENT	1ST HARMONIC	3RD HARMONIC	5TH HARMONIC	7TH HARMONIC
A	0.413	1.239	2.065	2.890
B	0.413	1.239	2.065	2.890
C	0.685	2.050	3.420	4.800
E	0.725	2.175	3.620	5.070
F	1.873	1.915	1.995	2.104
G	0.110	0.323	0.518	0.688
H	0.261	0.087	0.052	0.037
K	1.077	0.359	0.215	0.154
L	0.147	0.049	0.029	0.021
M	0.777	0.259	0.155	0.111

setting it up and checking it. In fact, it was possible to write down the coefficients without making current balances at all.

The principal characteristics will be described in conjunction with Table 1 and Fig. 6. In the table, junction points in Fig. 6 at which current balances are to be made are listed along the top. The unknown *a* and *b* components (real and imaginary, respectively) of the temperature vector are listed beneath each point. Row numbers 1–18 are listed along the left margin and column numbers 1–18 along the bottom. The unnumbered column at the extreme right contains known heat flow input values which are to the left of the *equals* sign when written in equation form. For example, row 6 represents the equation:

$$-1.43b_m + 2.16b_h - 0.12b_l - 0.17b_p - H = 0$$

To conserve space, only the first 3 digits of each coefficient are given in Table 1; the more precise values actually used in the solution appear in Fig. 6. Letters such as *A* and *B* refer to coefficients whose values are different for each harmonic; *i.e.*, coefficients which are frequency dependent. Numerical values are listed in Table 2.

The order in which junction points are listed at the top (k, m, h,) is not important, but it is important that current balances be made around each point in the same order. When this procedure is followed the matrix of coefficients is symmetrical about its principal diagonal. This diagonal is formed by an imaginary straight line connecting the arrowheads in the upper left and lower right. *Symmetrical* means that the same number occurs in corresponding blocks on either side of the principal diagonal. For

example, the row 5, column 17, value is +0.17. Interchanging rows and columns, +0.17 should also appear in row 17, column 5, which it does. This, then, is one check on all elements of the matrix except those along the principal diagonal.

All elements along the principal diagonal have the same row and column numbers. When a number lies in a block along the diagonal, considerable information can be obtained. Consider the element, row 6, column 6, in which appears the number +2.16. Since it lies along the diagonal and under b_h of point h, this equation is a result of the current balance around point h. Moreover, since it appears under b_h, it is the imaginary part of the equation. The real part is evidently the equation along row 5 because the element along the diagonal of the matrix and under a_h is in that row.

Further scrutiny will disclose that only real parts occur along odd numbered rows and only imaginary parts along even numbered rows.

Starting in the upper left and progressing down the diagonal, each value is seen to occur twice, first negative and then positive. The signs are therefore easily checked. Furthermore, each number represents the sum of all admittances around the point indicated at the top of the column. For example, −4.65 at row 7, column 7, is the sum of all admittances about point f in Fig. 6. Values along a row but not on the diagonal represent the admittance between the point indicated at the top of its column and the point around which the current balance is being made. For example, in row 7, column 15, is found 0.05. Column 15 is under point b. The element along the principal diagonal (row 7, column 7) is under point f. Therefore, 0.05 is the admittance between points b and f. Reference to Fig. 6 shows the first 3 digits are indeed 0.05.

It may seem strange that point c does not appear in Table 1. The conditions of the ASHAE problem are that point c is always maintained constant at 75 F, which is also the room air temperature. Hence, point c is a junction at which the temperature is known; therefore, it does not appear in the list of unknown temperatures in Table 1.

Letters A, B, C, and E in the matrix represent capacitors in parallel with resistors. They are always preceded by a negative sign and take positions under their points in blocks just to the right and left of the principal diagonal in the odd and even rows, respectively.

Only under point p does a frequency dependent coefficient occur along the principal diagonal. Inspection of Fig. 6 reveals that in summing the admittances around point p a resistance-capacitance series branch is involved. Since the capacitor is frequency dependent, the total admittance value is likewise frequency dependent. Values for F are included in Table 2.

By taking advantage of regularities in the matrix of coefficients, it was possible to eliminate considerable labor in writing the equations and, even more important, develop a checking procedure minimizing the possibility of errors.

The limitations of the technique used by Nottage must also be recognized. With respect to developing application engineering data, the most serious is the inability to handle other than periodic heat flow problems. For cooling equipment that is operated 24 hr a day, the technique can be used to develop a considerable amount of data, as will be shown subsequently. However, many air-conditioning systems are shut-down overnight. This mode of operation results in a rise in room air temperature followed by an early morning pull-down with the refrigeration equipment. Intermittent equipment operation introduces transient heat flow conditions that are better handled by analogue computers or numerical techniques such as the popular Schmidt[9] method.

Another limitation is that a linear circuit must be assumed, which means that resistors and capacitors are independent of temperature or heat flow. It is known that inside film coefficients, for example, are a function of temperature difference. Whether or not such nonlinearities have serious influence on thermal circuit solutions remains to be proved.

Additional thermal storage can be made available by allowing the room air temperature to vary. When a specified variation is impressed on a thermal circuit, the technique used by Nottage is easily applied. However, when the variation in room air temperature is the unknown, the technique used by Nottage becomes complex. Unfortunately, the latter condition is of more practical concern.

A study of typical configurations and conditions for which application engineering information is desired indicates an analogue computer capable of handling non-linear problems will be the eventual replacement for the technique used by Nottage.

DEVELOPING APPLICATION ENGINEERING INFORMATION

In developing information for application engineers to use in accounting for thermal storage in structures, considerable attention must be given to the manner in which the information is presented. In the field, time is of the essence. Therefore, all data must be in the simplest possible form so equipment can be quickly selected. Moreover, all buildings differ in some respects so the information must be extremely general in nature. A few of the variables are: (1) lighting load, (2) percent glass area, (3) type of window shading, (4) direction that each outside wall faces, (5) structural materials, and (6) outdoor weather conditions during a design day.

An added complexity is that thermal storage influences the equipment load during all hours of the day, not just during the time of the peak heat gain. Referring to Fig. 3, the cooling load is less than the heat gain until 2 p.m., after which time the load is greater than the heat gain. At 4 p.m. the load on this south exposure is double the calculated heat gain. Buildings usually have more than one exposure, *i.e.*, one outside wall may face south and another west. Thus, in determining the maximum load that the central station equipment must handle, and the time this load occurs, heat gain *vs.* cooling load relationships for *all exposures* must be considered. The relative area of each exposure also influences the load calculations.

Considering all the combinations of variables that are involved, it is evident that developing curves similar to Fig. 3 for each condition would be an impossible task. Therefore, methods had to be found for developing the information with a reasonable expenditure of time and money.

Study disclosed that the presumed linear nature of the circuit results in considerable simplification. First of all, each input can be applied separately to a circuit and the results added to determine the total effect. After selecting a typical office configuration and construction, the effect of outdoor temperature cycle, solar radiation, and lights can be determined individually. A series of temperature cycles typical of geographical regions can then be impressed separately and the circuit solved. Solar radiation inputs for East, Southeast, South, and so on, and 2 or 3 latitudes add to the data obtained. Then, for a particular region, appropriate temperature and solar radiation results are merely added together.

For different lighting intensities, the procedure is simple. Presume 3 watts per sq ft are impressed on a structure over a specified period of time and the cooling load *vs.* time curve is found. A lighting load of double this amount (6 watts per sq ft) merely doubles the previous load value at every instant of time. Stated differently, the *6-watt curve* is twice as high as the *3-watt curve*. Similarly, a 2-watt per sq ft curve would be two-thirds the height of the 3-watt curve.

A change in the level of the room temperature being maintained changes steady-

state values (areas under the load-time curves) but not the shapes of the curves. Hence, a load curve for an 80 F room temperature will be below that for a 75 F room temperature by an amount that is easily calculated.

Additional simplification came about by studying the influence of certain variables on circuit solutions. Some were found to be negligible while others, such as construction materials and different combinations thereof, were radically reduced in number. For example, an important variable is the percent of gross outside wall that is glass. Presume a load curve for a typical office has been determined for the condition of 30 percent glass. The load curve for 60 percent glass area is sought. The total solar radiation entering the space is double the 30 percent condition. However, the intensity per square foot of total interior absorbing surface is greater than double because the outside wall area has been decreased by the area of the additional glass. This effect was evaluated by studying the circuit diagrams and equations, as well as solving selected circuits and comparing results. The conclusion was that if the original circuit included a median amount of glass, then variations above and below had negligible effect. The significance is that linear combinations can be used; *i.e.*, the load curve for 60 percent glass can be taken as twice that for 30 percent glass.

Utilizing the experience gained in thermal circuit analysis, application engineering data for a variety of configurations and conditions, and 24-hr equipment operation, is being developed by means of the technique used by Nottage.

Several difficulties are yet to be overcome. Procedures for handling either variable room temperature or shorter daily operation of refrigeration equipment are not yet completely developed. Thermal conductivity, density and specific heat data for many materials are inadequate. Influence of nonlinearities, although believed insignificant, needs to be firmly established. Finally, it eventually must be determined by collection of field test data whether or not thermal circuit solutions correctly predict the behavior of actual structures.

CONCLUSIONS

1. The technique used by Nottage for thermal circuit analysis is a practical method of developing application engineering data, provided economical methods are available for solving large sets of simultaneous equations.

2. Thermal circuit analysis is a valuable tool for analyzing complex heat flow problems. Experience shows that engineers become adept at applying it after a small amount of training.

3. Analogue computers or numerical procedures appear more feasible than the technique used by Nottage when transient, rather than periodic, heat flow problems are encountered.

ACKNOWLEDGMENTS

Acknowledgment is made to the following of the Carrier Corporation Research Department: Mrs. Anna D. Mathill, mathematician, for developing supplementary 650 computer programs; Gaylord O. Ellis, senior research engineer, for devising the special relaxation procedure; and the mathematical analysis group for performing the computations.

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DISCUSSION

H. BUCHBERG†, Los Angeles, Calif. (WRITTEN): The authors and the company they represent should be commended for the effort being made to develop more accurate methods of system analysis and load prediction. Unfortunately the writer received this paper for discussion on June 20th which did not permit time for a detailed reading. Therefore, the discussion may appear rather general in nature.

1. The method of solution of the thermal network used by the authors required the solution of many simultaneous algebraic equations which was found to be quite time consuming. The time required was apparently reduced by forming a matrix and solving the problem on a 650 computer. In the work done at the University of California it was found that the simplest and most rapid steady state sinusoidal solution is achieved with an a-c network computer that gives the transfer function directly without the necessity of setting up and solving simultaneous equations. The transfer functions characteristic of typical constructions can be pretabulated for field use. The use of a simple graphical procedure for summing the contributions of all wall, roof, and floor sections computed from pretabulated transfer functions is now being written up in a paper that will be presented in the near future.

2. It is agreed that one of the limitations of the steady state sinusoidal method is the difficulty of handling problems where transients are important. For these cases the transient response method proposed by W. R. Brisken and co-workers has a definite advantage. Another limitation of the steady state sinusoidal method is the difficulty of handling the inside radiation exchange network for cases where radiation exchange on the inside of the room is important. This kind of problem and one in which nonlinearities exist in other parts of the network is best solved on a d-c network computer demonstrated by the writer in a previous paper. Steady inside radiation sources, however, can be handled with the steady state sinusoidal method.

3. One further comment in regard to the importance of input harmonics might be of interest. The assessment of the importance of input harmonics in load calculations

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can sometimes be misleading when examining the contributions of single wall sections. Very often use of the fundamental input only is sufficient because of the small contributions made by many structural sections. This will be demonstrated in a paper now almost completed for publication.

AUTHORS' CLOSURE (Mr. Clausen): The authors wish to thank Professor Buchberg for his commendations and for his comments.

The authors are in agreement that the time required to solve a specific problem may be less when utilizing an a-c network computer. However, because of the systematic pattern of the heat balance equations, they have found that computer personnel soon learn to write these equations for complex problems. This releases the engineer for defining the problem and analyzing the results, which must be done in any case. For the calculation of application data, a general thermal circuit for the desired building geometry could be set up and the equations written in general form. Varying the factors affecting the problem such as construction, film coefficients and loads, would mean only changing a few values in the matrix and re-solving the set on the machine.

A major limitation of the technique used by Nottage is that it is unable to handle transient heat flow problems, as reemphasized by Professor Buchberg. As stated in the paper, analogue computers or numerical procedures appear more feasible than the method used by Nottage for transient heat flow problems.

1606

No. 1606

THERMAL PERFORMANCE OF FRAME WALLS

PART III—WALL SURFACE TEMPERATURES

By G. O. HANDEGORD*, SASKATOON, CANADA

SELECTION of insulation for use in frame walls under cold weather conditions is often based solely on the calculated overall thermal properties of the resultant wall section. This approach may be adequate from the standpoint of heating system design or interior comfort evaluation where overall average values are involved. However, it may not always provide an accurate appraisal of wall performance in regard to such problems as surface condensation or dust marking.

Two previous papers[1,2] demonstrated that actual surface temperature patterns exhibited by frame walls may differ widely from those predicted on the basis of simple theory. Both papers emphasized the vertical variations in temperature caused by convective action in the wall space. The second paper[2] provided some additional information on horizontal surface temperature variations resulting from lateral heat exchange at studding. It was suggested that the surface temperature gradient in the vicinity of studding was largely influenced by the arrangement of insulation in the stud space and by the degree of thermal protection afforded the sides of the stud.

The amount by which surface temperatures over studs are lower than surface temperatures between studs may be important in connection with the possible occurrence of condensation on the inside surfaces of insulated frame walls. The studies of vertical temperature variations referred to above have shown that, at low outside temperatures, inside surface temperatures near the bottom of such walls may be lowered to the point where condensation is imminent even at moderate relative humidity. For this reason, it may be desirable to select insulation arrangements that lead to the highest possible surface temperatures over studs.

Steep surface temperature gradients will normally result in *dust marking*. This phenomenon, discussed in detail by Nielsen[3], is due to differences in the rate of accumulation on surfaces at different temperatures. The seriousness of the prob-

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[1] Exponent numerals refer to References.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Murray Bay, Que., Canada, June 1957.

lem is dependent upon the period necessary for a noticeable pattern to appear and the relationship of this period to the normal redecoration schedule. Walls that exhibit severe temperature gradients in the vicinity of studding will require a shorter period of time for noticeable patterns to form and it is therefore desirable to select and locate insulations within the wall to provide the most uniform surface temperatures possible.

The study reported in this paper was undertaken to obtain more complete information on the horizontal surface temperature patterns resulting from the use of different types and arrangements of insulation in a conventional frame wall. Consideration was given to suitable methods for prediction of surface temperature patterns on frame walls with particular emphasis on walls with semi-thick batt or blanket insulation. The project was undertaken as part of a program on Cold Weather Wall Research at the Prairie Regional Station of the Division of Building Research, National Research Council of Canada, at Saskatoon.

TEST APPARATUS

The test apparatus has been described previously[1]. It consists basically of a cold-room—warm-room combination in which a full-scale frame panel can be subjected to a temperature gradient under controlled conditions. Regulation of air temperature on either side of the panel was within $\pm\frac{1}{2}$ F. deg. Forced circulation of air was maintained upward over the cold face of the panel during all tests; gravity circulation was employed in the warm room.

All temperature measurements were carried out with 30 B & S gage, copper-constantan thermocouples in conjunction with an electronic indicating potentiometer. Surface thermocouples were cemented in grooves so as to be flush with the wall surface. At least 2 in. of the lead wire from the thermojunction was similarly secured to the wall. Thermocouples were located at points midway between studding, on the centerline of the stud, and 2 in. from the stud centerline at 3 vertical positions on the wall section. Additional thermocouples were installed at various locations throughout the wall.

Heat meters were used in most of the tests to determine rates of heat flow into areas over the stud and midway between studding. These meters were of the multiple differential thermocouple type, with the sensitive elements mounted between synthetic resin sheets $4\frac{1}{2}$ in. square. Each meter was held against the wall surface by 4 spring-loaded polystyrene rods which extended from a carriage that could be moved vertically up the wall. The emf generated by the thermopile was recorded for a period of at least 1 hr at each location using an electronic strip chart potentiometer.

A third heat meter was used in one of the tests to provide a measure of the rate of heat flow into the center of the sheathing at mid-height. This meter was secured to the warm face of the sheathing with $\frac{1}{4}$ in. No. 6 wood screws.

TEST PROCEDURE

The same basic wall panel, 8 ft in height by 5 standard stud spaces in width, was used for all tests. The two outer stud spaces were insulated with mineral wool insulation and were left unchanged during all studies. The different types of insulation tested were installed in the 3 inner spaces, but all measurements were confined to the center space.

Modifications to the insulation of the test panel were made prior to weekends with the subsequent test period being of 3 to 4 days duration the following week.

Four complete temperature traverses, and 2 complete heat meter traverses were made during this time.

DESCRIPTION OF WALLS TESTED

The basic wall panel consisted of 6- by ½-in. bevel cedar siding as exterior finish, sheathing paper, 25/32- x 10-in. spruce shiplap sheathing, 2- x 4-in. framing (actual dimensions 1½ x 3½ in.) and ⅜-in. gypsum wallboard. The gypsum wallboard was secured to the framing with 1-in. No. 12 wood screws so that it could be removed to change the insulation.

Twenty-six different insulation arrangements were tested involving both reflective and mineral wool types. Six forms of commercial house insulation were studied and these were applied as carefully as possible according to the manufacturers' directions. The commercial insulations studied are best illustrated by the horizontal sections shown in Fig. 1. For further tests on insulation fit at the stud, a preformed low-density, glass fiberboard-type insulation was cemented to the cold side face of the plasterboard and cut to accommodate the framing. The insulation and plasterboard were then removed as a unit and the insulation edges recut to the required shape at the studding for each succeeding test.

DISCUSSION OF RESULTS

Relative Performance of Different Insulations: Fig. 1 provides a summary of the results obtained on 11 wall constructions having different types and arrangements of insulation. The values listed represent the average of measurements taken at the 2-, 4-, and 6-ft levels on each wall. Column 2 shows the cross-sectional details of each panel together with the associated horizontal surface temperature pattern. The plotted curves are based on readings taken over one-half the stud space width. Symmetry has been assumed in extending the curve for the width shown.

All panels were tested under similar conditions with warm-side air temperatures at approximately 70 F and cold-side air temperatures at approximately —30 F. The warm-side air temperatures are listed for each wall construction in Column 3 and the inside-to-outside air temperature difference is shown in Column 4. The greatest departure from the test conditions occurred for walls Nos. 5 and 9, where the inside-to-outside air temperature differences were 104.8 F and 98.3 F, respectively. The average differences in surface temperature between studding and over the stud are listed in Column 5 and the ratio between the rate of heat flow into the area over studding to that midway between studs is shown in Column 6.

Wall No. 5, in which the insulation consisted of 2 air spaces each faced one side with foil had the most uniform wall surface temperature pattern of any of the walls tested. This wall represents a case where the thermal resistance of the portion between studding is similar to that at the studs, and a close approximation to a thermally homogeneous wall results. In regions of severe winter climate, however, the thermal resistance provided by such a wall construction is inadequate from the economic standpoint and is substandard in regard to optimum interior comfort.

Of the walls in which reflective insulation was used wall No. 1 showed the greatest horizontal variation in surface temperature. In this wall the insulation provided in the stud space was basically 3 air spaces, 2 faced 1 side and 1 faced 2 sides with aluminum foil. In this respect, the insulation was equivalent to that in walls Nos. 2, 3, and 4, but these walls showed less severe variations in surface temperature.

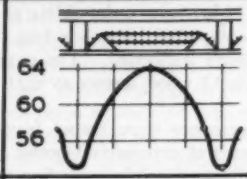
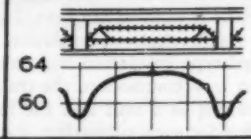
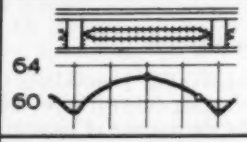
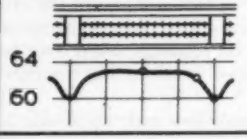
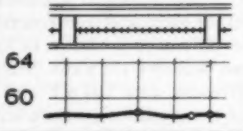
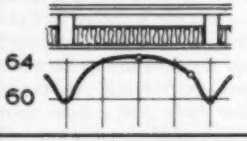
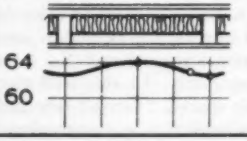
1	2	3	4	5	6	7	8
WALL NO.	TEMPERATURE PATTERN	T_i	$T_i - T_o$	ΔT	$\frac{Q_{st}}{Q_{ss}}$	ΔT	$\frac{Q_{st}}{Q_{ss}}$
① FOIL BLANKET		70.2	100.1	11.1	2.7	2.4	1.4
② FOIL BLANKET		70.2	99.9	4.9	1.5	2.4	1.4
③ FOIL BLANKET		69.9	100.0	4.0	1.6	2.4	1.4
④ TWO FOIL CURTAINS		69.9	99.9	2.8	1.5	2.4	1.4
⑤ ONE FOIL CURTAIN		70.7	104.8	0	–	0.5	1.1
⑥ TWO INCH BATT		70.4	100.4	4.9	1.7	2.8	1.5
⑦ TWO INCH BATT		70.4	100.4	1.5	1.1	2.7	1.5

FIG. 1—SURFACE TEMPERATURES ON FRAME WALLS WITH DIFFERENT TYPES AND ARRANGEMENTS OF INSULATIONS

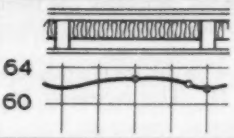
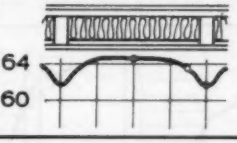
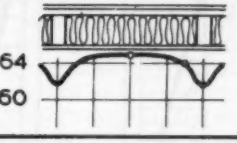
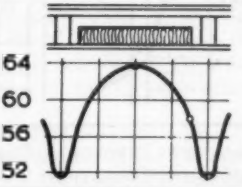
1	2	3	4	5	6	7	8
WALL NO.	TEMPERATURE PATTERN	T_i	T_i-T_o	ΔT	$\frac{Q_{st}}{Q_{ss}}$	ΔT	$\frac{Q_{st}}{Q_{ss}}$
(8) TWO INCH BOARD		69.5	98.9	1.1	1.1	2.7	1.5
(9) THREE INCH BOARD		69.8	98.3	3.1	1.7	4.0	2.0
(10) ROCK WOOL FILL		69.9	100.3	4.0	1.8	4.3	2.1
(11) TWO INCH BOARD		69.5	99.6	12.0	3.5	2.8	1.5

FIG. 1 (*continued*)—SURFACE TEMPERATURES ON FRAME WALLS WITH DIFFERENT TYPES AND ARRANGEMENTS OF INSULATIONS

Examination of the cross-sectional details for these 4 walls shows that this improvement in surface temperature pattern is at least partially associated with the degree of exposure of the sides of the studding. In wall No. 1 the stud was exposed to the cold outer air space for its entire depth; in wall No. 2, for two-thirds of its depth; in wall No. 3, for half its depth, and in wall No. 4 only one-third of the side of the stud was so exposed. The reflective properties of the insulation tab in walls Nos. 2 and 3 would also tend to increase the surface temperature difference by decreasing the lateral flow of heat into the sides of the stud from the warm-side air space.

The effect of insulation location may also be noted from a comparison of walls Nos. 6 and 7, where identical batt type insulation was installed. The outer portions of the studs in wall No. 6 were exposed to the low air temperature of the space, while in wall No. 7 the entire stud was in a region of much higher temperature. The lateral heat flow from the sides of the stud in wall No. 6 and the rate of heat

flow into the face of the stud would be greater, with a consequent lowering of surface temperature over the stud.

Wall No. 8 was essentially the same as wall No. 7, except that a slightly thicker and more uniform mineral fiber insulation was installed. It is included for comparison with wall No. 9 to illustrate the effect of increasing the thickness of insulation in the stud space. As can be noted, increasing the thermal resistance of the stud space will tend to increase surface temperature difference. Wall No. 10 is a further example of this effect.

The marked influence of lateral heat flow on surface temperature pattern is again apparent from the results for wall No. 11. This represents an example of poor

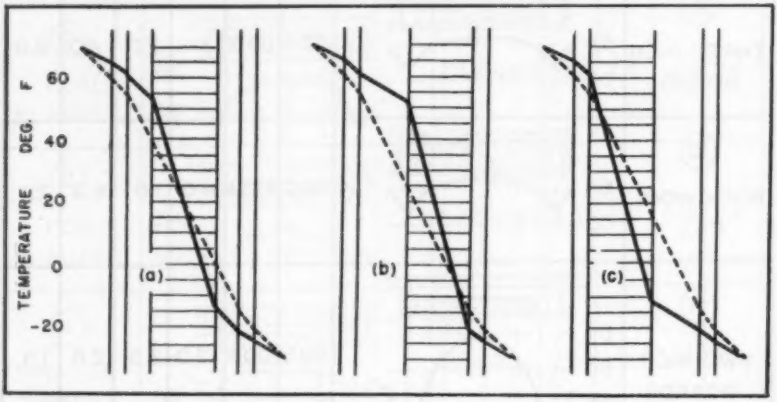


FIG. 2—CALCULATED TEMPERATURE GRADIENTS THROUGH A FRAME WALL WITH DIFFERENT LOCATIONS OF INSULATION

insulation application and will be discussed later. It is included here as a case of extreme horizontal surface temperature difference.

Comparison of the results presented in Fig. 1 indicates that the lateral flow of heat within frame walls may have an appreciable effect on the warm-side surface temperature pattern. The location of semi-thick insulation within the wall has a marked effect on surface temperature as does the fit between the insulation and studding.

Certain general principles of frame wall design for uniformity of surface temperature are suggested by these results. Specifically, semi-thick batt or blanket insulations should be installed as close to the cold side of the space as practicable and should provide maximum thermal protection to the sides of the studding.

Since horizontal surface temperature gradients have implications with regard to the occurrence of both dust patterns and condensation on inside surfaces, the assessment of walls with respect to these gradients is highly desirable. Limiting criteria for surface temperature difference with regard to dust marking have been suggested by some authorities[4,5] and have been applied to building regulations.[4]

Evaluation of Wall Surface Temperature Differences: The most direct way of determining wall surface temperatures is by actual testing, but the cost of such testing for the variety of constructions possible is prohibitive. Rigorous mathematical analysis, on the other hand, is difficult if not impossible because of the extremely complex nature of the heat flow pattern. However, for certain types of wall constructions, calculations on the assumption of parallel heat flow paths may give reasonably accurate results.

The surface temperature difference and ratio of heat flow in at the stud to that midway between studding, calculated on the assumption of parallel paths for the walls in Fig. 1, are listed in Columns 7 and 8. These values were determined on the basis of data given in THE GUIDE of ASHAE.

Reasonably close agreement between calculated and observed values of surface temperature differences and heat flow ratios may be noted for walls Nos. 4, 5, and 10. Examination of the cross-sectional details of these walls will show that, in each case, the insulation was of uniform thickness across the entire width of the stud space and was symmetrically placed in relation to a plane parallel to the wall at the center of the air space. In these 2 respects they differed from the other walls tested.

This agreement between calculated and observed values may be partially explained by temperature gradients shown in Fig. 2. In Fig. 2 (a) the calculated temperature gradients through the stud and through the stud space are shown superimposed for a frame wall with insulation located at the center of the air space. In Fig. 2 (b) the same insulation is placed at the outside, and in Fig. 2 (c) at the inside of the space.

With insulation at the center of the air space [Fig. 2 (a)] the temperature levels imposed on the sides of the stud are such that the lateral flow of heat into the stud near the warm side is balanced by the lateral loss near the cold side. The rate of heat flow into the stud will therefore approach that for a condition of no lateral heat exchange.

Location of insulation as in Fig. 2 (b) will expose the sides of the stud to a higher temperature regime and the additional lateral heat gain will reduce the rate of heat flow into the face of the stud. The resultant surface temperature over the face of the stud will thus be higher than that calculated by simple theory, as it was for wall No. 7.

The opposite condition will occur with insulation located as shown in Fig. 2 (c): the sides of the stud will be exposed to a lower level of temperature, increased heat loss from the sides of the stud will occur, and a lower surface temperature than that predicted will result. This condition is exemplified by the results for wall No. 6.

The interior and exterior cladding materials of this wall section have been assumed to possess identical thermal characteristics. This is not strictly true for the walls tested, but the same general relationship between temperature gradients would exist. This simple analysis, when considered with the test results, would indicate that reasonably accurate values of surface temperature difference for frame walls with plaster finish may be calculated on the assumption of parallel heat paths when insulation is installed at approximately the center of the air space of the wall and when the insulation is of uniform thickness across the width of the stud space.

The calculation of surface temperature difference using the foregoing procedure may also be employed as a basis for qualitative assessment of walls in which the insulation is installed in the cold side of the space. The calculated value in such

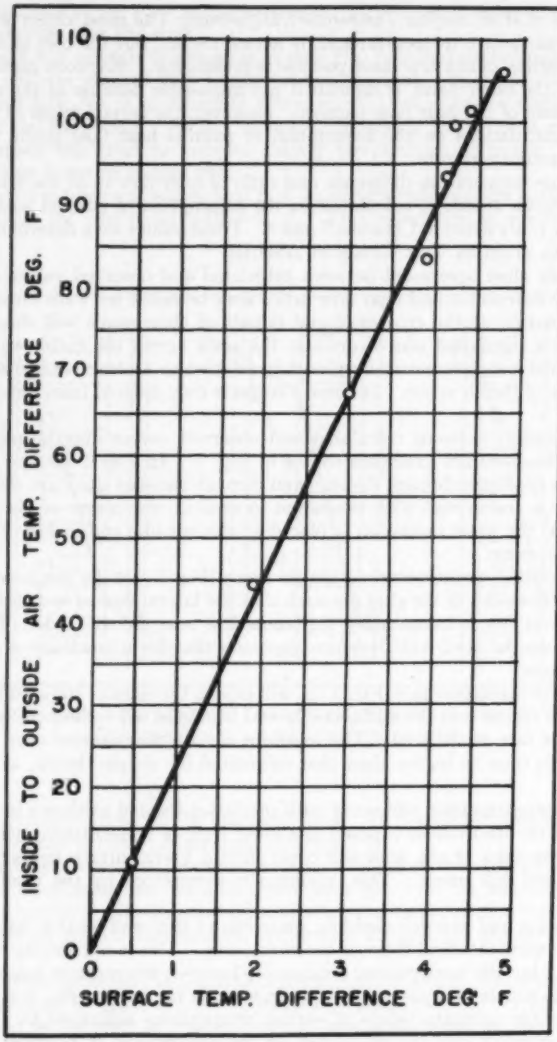


FIG. 3—SURFACE TEMPERATURE DIFFERENCE AS RELATED TO OVERALL AIR TEMPERATURE DIFFERENCE

cases represents a maximum, and provided this does not exceed the accepted criterion, the wall may be regarded as acceptable.

The assessment of walls in which the insulation is installed in the warm side of the space is more difficult, first because a limit cannot be established by simple computation, and secondly because the shape of the insulation at the stud has a marked influence on the surface temperature pattern. For a specific type of wall construction and insulation, however, test data may be used as a basis for assessment, or at least for evaluation of limiting conditions.

The majority of houses in Western Canada are of reasonably standard frame construction; the most widely used insulation is of the mineral wool or glass fiber batt type, nominally 2 in. thick. The most common wall construction is of the type shown for wall No. 6 in Fig. 1. The maximum surface temperature difference to be expected with 2-in. batt insulation may, therefore, be assumed as 4.5 F when the inside-to-outside temperature difference is 100 F, provided the insulation is of uniform thickness across the stud space. For walls with this type of insulation installed between this position and the center of the air space a value may be assumed as between the limits of 4.5 F and the calculated value of 2.8 F, under the same ambient conditions. When the design temperature difference is other than 100 F proportionality may be assumed. This assumption of proportionality is inherent in the calculation procedure and is reasonably correct for walls such as No. 6, as shown by the test results in Fig. 3. The plotted points were obtained for wall No. 6 at different inside-to-outside air temperature differences ranging from 10 F to 104 F.

Insulation Fit at Studding: These procedures for calculating wall surface temperatures are dependent on the insulation being of constant thickness across the entire width of the stud space. Many batt and blanket type insulations do not have a truly rectangular edge profile, and in some cases may have a cross-section approaching that of the reflective insulation in wall No. 1. Also the spacing of studding in actual construction does not always coincide with the 16-inch module, and in some cases a condition analogous to that of wall No. 11 may occur. A survey of houses in Canadian cities in 1951 indicated that 5 percent of the wall area, exclusive of window and door openings, had stud spacings which were greater than 16½ in.[6]

To evaluate the relationship between insulation fit at studding and surface temperature variation, 3 series of tests were conducted using a uniform, light density, glass fiber insulation, 2 in. thick, which was cut to provide varying degrees of exposure of the stud. The main purpose of this study was to provide a basis for rating batt insulations in regard to surface temperature patterns in terms of their measured profile, without recourse to actual testing.

The results of these 3 series of tests are shown in Figs. 4, 5 and 6, in which the difference in surface temperature between locations over the studding and between studding is plotted against the cross-sectional area of the space between insulation and stud. Corresponding rates of heat flow into the area over studding and between studding are also shown in these figures, and in Fig. 5 the rate of heat flow through the sheathing at mid-height of the wall is included. The conditions of test were identical in all cases, warm-side air temperatures being maintained at approximately 70 F, with cold-side air temperatures at approximately —30 F. The plotted points represent average values taken at the 2-, 4- and 6-ft levels, except for the heat flow through the sheathing in Fig. 5.

The effect of chamferring the insulation at various angles to the stud is shown by the results of Series I tests in Fig. 4. Apparently the average width of the space

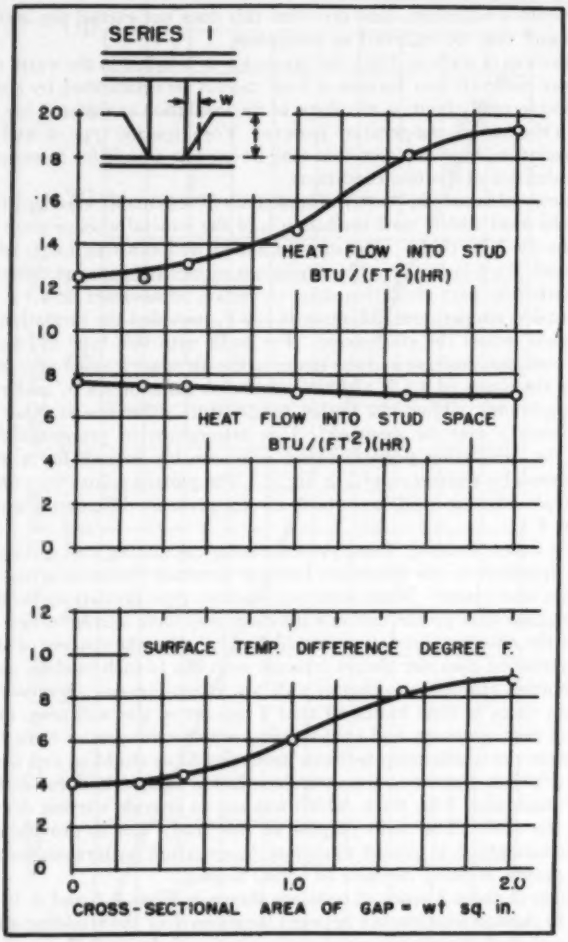


FIG. 4—SURFACE TEMPERATURE DIFFERENCE AND HEAT FLOW RATE AS RELATED TO INSULATION SHAPE—SERIES I

between insulation and studding is most critical in the range from ¼-in. to ¾-in., as indicated by the shape of the curves for temperature difference and heat flow into studding. It will be noted that the rate of heat flow into the center of the stud space decreased slightly with increasing stud exposure. This decrease was no doubt due to the increase in temperature of the air space brought about by increased lateral heat flow from the studs.

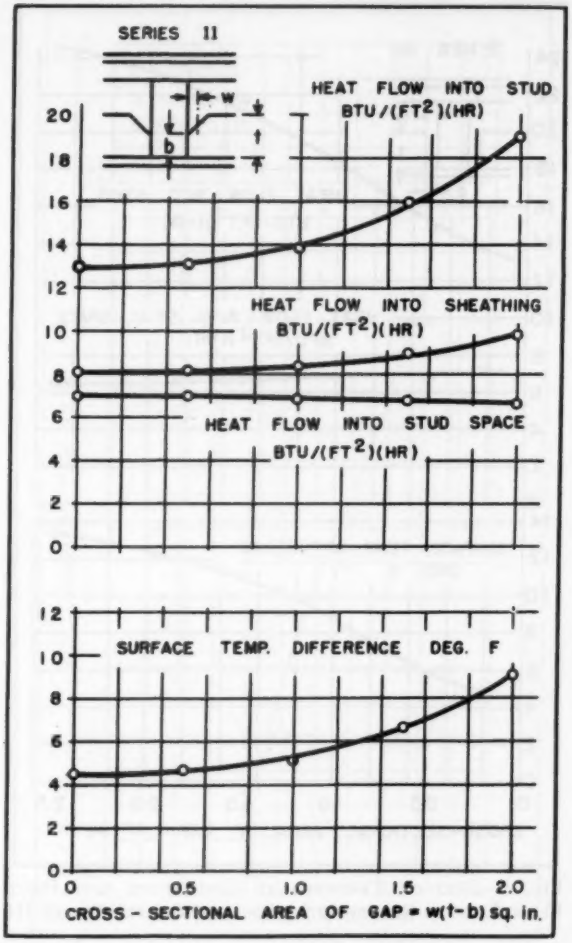


FIG. 5—SURFACE TEMPERATURE DIFFERENCE AND HEAT FLOW RATE AS RELATED TO INSULATION SHAPE—SERIES II

The results of chamferring the insulation at 45 deg to the stud to provide different widths of stud exposure are shown by the Series II tests in Fig. 5. The shape of the temperature difference and heat flow curves in this figure are different from those of Series I, the rate of change of surface temperature difference increasing with increasing cross-sectional area of gap. The extreme case in Series II was identical to that of Series I and reasonably close agreement between results was obtained.

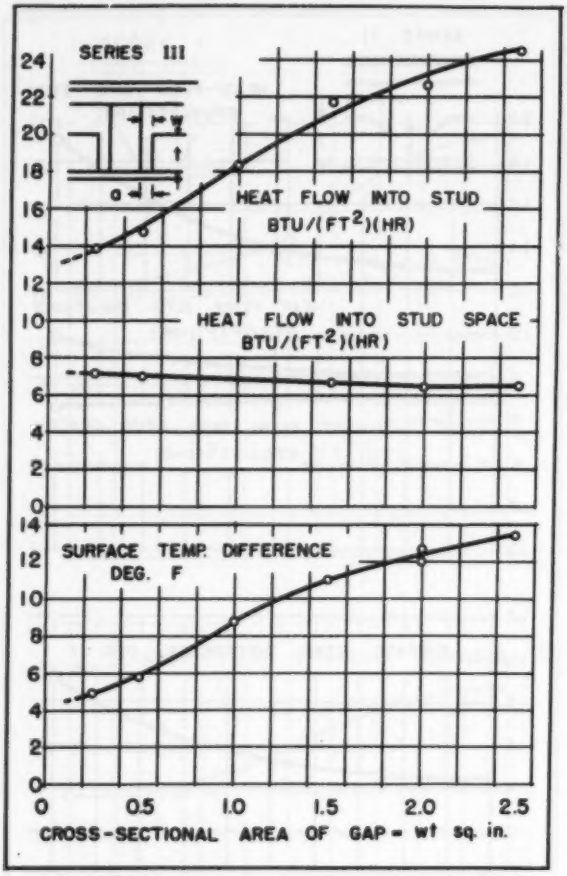


FIG. 6—SURFACE TEMPERATURE DIFFERENCE AND HEAT FLOW RATE AS RELATED TO INSULATION SHAPE—SERIES III

For all practical purposes a mean temperature curve for the results of both Series I and Series II might be employed since the individual experimental points would fall within 1 F deg of such a curve.

Further evidence of the lateral flow of heat from studding to the air space and through the sheathing is provided by the heat flow measurements of Fig. 5. The rate of heat flow into the wall between studs decreased, while the rate of heat flow through the sheathing increased with increasing stud exposure.

The insulation cross-sections of Series I and II approximate the general shape of some batt and blanket insulations as installed in framing which is spaced at 16 in. or less on centers. Occasionally, however, stud spacings in practice exceed this

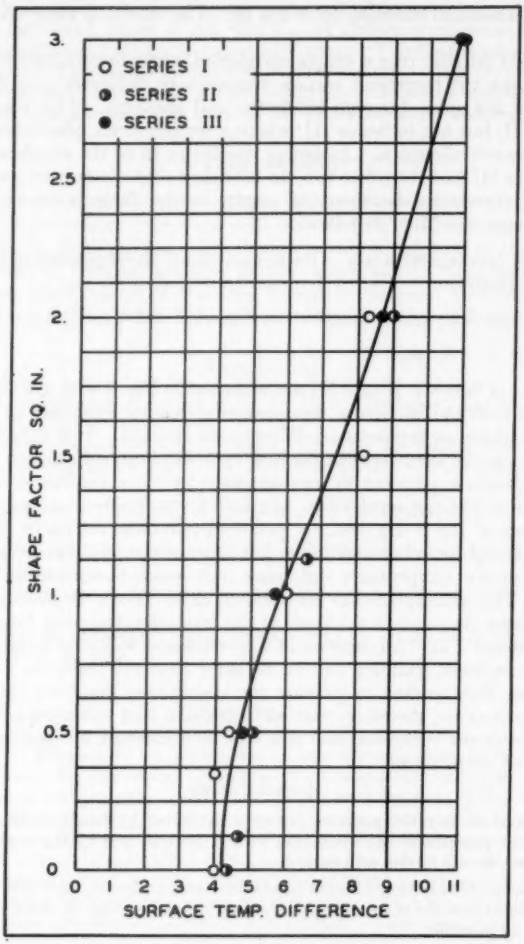


FIG. 7—SURFACE TEMPERATURE DIFFERENCE (IN PERCENT OF INSIDE TO OUTSIDE TEMPERATURE DIFFERENCE) AS RELATED TO SHAPE FACTOR FOR FRAME WALLS WITH NOMINAL 2-IN. BATT INSULATION

modular value and such situations must be recognized in the assessment of this type of insulation.

Series III tests were undertaken to obtain some additional information on the surface temperature patterns with oversize stud spacing. Rather than change the width of stud space, however, the insulation was reduced in width to provide gaps

between insulation and studding up to 1¼ in. The results of these tests are shown in Fig. 6.

It was hoped initially that a simple correlation might be obtained for all 3 series of tests between the horizontal surface temperature difference and the cross-sectional area of the space between insulation and studding. This was the case in Series I and II, but not in Series III, where a section of the plasterboard was also exposed to the cold air space. Assuming this factor to be the significant difference between Series III and the other two, an arbitrary *shape factor* was used to provide an empirical correlation between the results of the three series of tests. This *shape factor* was calculated as follows:

Shape factor = (average thickness − thickness at stud) (average width of space + width of space at plaster).

Using the terminology of the diagrams in Figs. 4, 5 and 6,

Shape factor = $(t - b)(w + a)$.

The results of Series I, II and III are replotted in Fig. 7 with the shape factor as ordinate and with the horizontal temperature difference expressed as a percentage of inside-to-outside air temperature difference as abscissa. It is suggested that this relationship may be used to evaluate batt-type insulations in regard to horizontal surface temperature patterns by measurement of their profiles as installed in a mock-up wall. The test results were obtained for an inside-to-outside air temperature difference of 100 F deg, but, for practical purposes, the curve given in Fig. 7 may be employed for other conditions if a linear relationship is assumed between horizontal surface temperature difference and inside-to-outside air temperature difference. This assumption has been shown to be correct for walls such as No. 6.

In some cases the average thickness of the insulation batt may be greater or less than the nominal 2 in. An increase in batt thickness will tend to raise the surface temperature between studding and at the same time will lower the temperature of the air space, thus tending to increase the lateral heat loss from the studs. It is reasonable to assume, therefore, that an increase in batt thickness will increase the surface temperature difference and this will be accounted for qualitatively in the calculation of *shape factor*.

CONCLUSIONS

1. Horizontal surface temperature patterns exhibited by frame walls are greatly influenced by the position of the insulation within the wall and by the degree of exposure of studding to the air in the wall cavity.

2. When semi-thick batt or blanket insulations are employed, more uniform horizontal surface temperatures are obtained with the insulation installed as close to the cold side of the space as possible.

3. It is most desirable from the standpoint of minimum horizontal surface temperature variation that semi-thick batt or blanket insulations which are to be located toward the warm side of the air space provide protection corresponding to a substantial proportion of their thickness at the studding.

4. Calculation of surface temperature differences on frame walls on the assumption of parallel heat flow paths may provide reasonably accurate values provided the insulation is of uniform thickness from stud to stud and is approximately centered in the confines of the space.

5. When the insulation is placed toward the outside of the stud space calculation of surface temperatures on the assumption of parallel heat flow paths results in higher values of surface temperature difference than actually occur.

6. When the insulation is placed in the warm side of the stud space adjacent to the inside finish, calculated values of the difference in surface temperatures over and between studs are substantially less than actually occur. The fit of the insulations at the stud has a marked effect on the surface temperature pattern. Data are presented which permit the prediction of horizontal surface temperature difference for batt and blanket insulations taking into account the shape of the insulation in the vicinity of the stud.

ACKNOWLEDGMENT

The author wishes to acknowledge with thanks the assistance of D. G. Cole in the taking and recording of the data used in this study. The comments and suggestions of Dr. N. B. Hutcheon and other members of the staff of the Division of Building Research, Ottawa, are also greatly appreciated. This paper is a contribution from the Division of Building Research, National Research Council of Canada, and is published with the approval of the director.

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DISCUSSION

M. W. KEYES, Pittsburgh, Pa. (WRITTEN): There are only 2 points which I wish to make in connection with this paper. These concern constructions which would doubtless provide interest and which were not included in the tests reported in the paper:

1. Insulation blanket applied over the studding, between the studding and the sheathing, with aluminum foil vapor barrier on the warm side of the insulation. This construction would put practically all of the studding on the warm side of the insulation. In addition the aluminum foil would tend to warm the studding with heat picked up from the warm side of the insulated space between the studs. This should make for a very uniform temperature gradient on the inside wall surface. This construction is beginning to come into use.

2. Semi-thick insulation applied as in wall No. 7 (against the sheathing) and with aluminum foil backed plaster board used as interior finish instead of the plain plaster board indicated in wall No. 7. It may be expected that the heat transmission through the aluminum foil from the wall surface between the studding to the area directly over the studding will maintain a more even temperature gradient across the inside wall surface.

S. F. GILMAN, Syracuse, N. Y.: This is an interesting paper and can perhaps be a source of additional and important information. Mr. Handegord has shown considerable test data on temperature differences between solid surfaces and the room air. Hence, if heat flow rates were also measured, it would be possible to evaluate the film

coefficients that existed during tests. With detailed knowledge of the heat flows, the variation in film coefficient as a function of temperature difference and type of material beneath the surface could be determined.

Film coefficient data in actual structures are very meager and represent an important stumbling block to accurate evaluation of thermal storage during transient and periodic conditions.

My question is, therefore, can any film coefficient data be extracted from these test results? If not, I urge that consideration be given to obtaining it in future tests since this is an excellent test facility and the data are urgently needed.

L. E. SEELEY, Westfield, Mass.: I have just a question in regard to the dust pattern. If dust collected where the temperatures were lower, and if there were wall paper on that wall, would one get the same effect?

AUTHORS' CLOSURE: The first construction referred to by Mr. Keyes would undoubtedly provide a fairly uniform horizontal surface temperature pattern. A somewhat similar wall was tested in our apparatus but not reported in this paper. The insulation in this wall consisted of a 2-in. glass fiber batt installed against the sheathing with a reflective foil surface on the warm side of the insulation. Surface temperature measurements on this wall showed a similar variation to those of Wall No. 7, but were all slightly higher due to the increased overall thermal resistance. Tests have also been conducted on a wall construction similar to the second one described by Mr. Keyes, but in this case horizontal temperature variations were greater than for the wall without reflective plasterboard. This effect was possibly due to reduction of radiant heat transfer from the outer face of the plaster to the sides of the studs, coupled with the increased thermal resistance of the section between studs, both of these effects being brought about by placement of the reflective surface on the warm side of the space. Tests on similar walls with 3-in. batt insulation showed no significant difference in horizontal surface temperature variation resulting from the location of the reflective surface. In all cases where a reflective surface is provided in the air space beneath the plaster a slightly greater vertical variation in surface temperature will be experienced, due to the predominance of convective heat transfer. In the types of walls mentioned by Mr. Keyes this vertical variation will not be serious.

The measurements made of the various wall constructions described in the paper could be used to calculate inside film coefficients, as Dr. Gilman suggests. The air flow conditions in the test room, however, may not be too representative of actual construction and the radiation environment would possibly be different. If it is felt that evaluation of the inside surface coefficients would be useful, even though peculiar to the apparatus, this information could certainly be made available. No measurements were made of outside surface temperature or rates of heat flow out of the wall and it would therefore not be possible to determine outside coefficients.

The question of the effect of wall paper on dust marking, raised by Dean Seeley, is one for which we have no test information. Our investigations have all been concerned with the potential for dust marking as determined by inside surface temperature variation. We have not conducted any experimental work to determine other factors relating to dust formation. It is quite likely, however, that certain interior finishes might prolong the time for noticeable marking to appear and, in this respect, the surface texture and colour might be somewhat significant. I feel that the thermal characteristics of wall paper would not have much influence on the temperature variation.

1607

No. 1607

SATURATION PRESSURE OF WATER ON THE NEW KELVIN TEMPERATURE SCALE

By JOHN A. GOFF*, PHILADELPHIA, PA.

THE SATURATION pressure of water is a thermodynamic property that both the air-conditioning engineer and the meteorologist want to know as accurately as possible. Goff and Gratch[1], in connection with their research under a cooperative agreement between the AMERICAN SOCIETY OF HEATING AND AIR CONDITIONING ENGINEERS and the University of Pennsylvania, have calculated this property from the very accurate calorimetric data of Osborne, Stimson and Ginnings[2,3] at the National Bureau of Standards, from available spectroscopic data, and from other relevant information regarding water. They have reported their results in the form of empirical formulas that were adopted as standard by the International Meteorological Organization in 1947[4] and by the Joint International Committee on Psychrometric Data in 1949[5]. They have also presented them in tabular form in the HEATING, VENTILATING, AIR CONDITIONING GUIDE[6] for the convenience of the air-conditioning engineer and in Smithsonian Meteorological Tables[7] for that of the meteorologist.

The saturation pressure of water is a pure temperature function. In the empirical formulas just referred to the argument of the function is absolute temperature T^* on the old Kelvin scale with $T_0^* = 273.16$ and $T_1^* = 373.16$ taken from Birge[8] as best values, subject to the stipulation that they differ by exactly 100, for assignment to the ice-point and steam-point isotherms, respectively. In the Smithsonian Meteorological Tables the argument is (thermodynamic) Centigrade temperature, $t^* = T^* - 273.16$, while in THE GUIDE it is the Fahrenheit temperature derived therefrom in the usual manner.

In October 1954 the Tenth General Conference on Weights and Measures[9] adopted a new Kelvin scale T of absolute temperature on which the triple-point isotherm, namely, the locus of states of a given substance in which the substance is in diathermic equilibrium with water in 3 coexisting phases (vapor, liquid, solid),

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* Professor of Mechanical Engineering, University of Pennsylvania. Member, Working Group on Physical Functions and Constants, Commission for Aerology, World Meteorological Organization.

[1] Exponent numerals refer to the References.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Murray Bay, Que., Canada, June 1957.

is assigned the value 273.16, exact by convention. The 2 scales deviate by appreciable if small amounts in the range of interest to the air-conditioning engineer and the meteorologist; hence it is desirable to transpose the Goff-Gratch formulas to the new scale T as argument, and the tables, to the new (thermodynamic) Celsius scale, $t = T - 273.15$. This is the object of the present paper.

The universal temperature function whose existence is asserted by the Second Axiom of thermodynamic theory is called absolute temperature because it has only 1 deg of arbitrariness, a multiplicative constant. This means that only the ratio of the absolute temperatures for assignment to 2 different isotherms can be determined experimentally. This ratio is usually determined by the methods of gas thermometry which, in principle, consist of 2 separate experiments. The first yields compressibility factor, $Z = pv/(pv)^0$, as a function of pressure p along each isotherm investigated, $(pv)^0$ being the limit that the pressure-volume product pv appears to approach as pressure tends toward zero along the isotherm. Burnett[10] has shown how to avoid volume measurements entirely in this experiment by so designing the apparatus as to place the observed pressures p_r ($r = 1, 2, \ldots$) at unobserved volumes $v_r = v_0 N^r$ in strict geometric progression with unobserved ratio N. The apparatus constant N can be inferred from the pressure sequence p_r as that which makes the quantity $N^r p_r$ appear to approach a finite value $(N^r p_r)^0$ as pressure tends toward zero along the given isotherm; then $Z = N^r p_r/(N^r p_r)^0$.

The second experiment required for the determination of the ratio of 2 absolute temperatures employs the constant-volume gas thermometer as apparatus. In it the pressure p of a gas whose compressibility factor has previously been determined in the manner just described is measured at the unobserved volume v on the T-isotherm, then its pressure p' at the same volume v' ($= v$) on the T'-isotherm is also measured. With the 2 values of compressibility factor, $Z = Z(p,T)$ and $Z' = Z(p', T')$, known, the ratio of absolute temperatures is computed from $T/T' = pZ'/p'Z$ because this puts absolute temperature directly proportional to the zero-pressure product $(pv)^0$ as required by thermodynamic theory. The important point to note from this brief exposition is that the ratio of absolute temperatures is determinable from measurements of pressure only along the 2 isotherms in question and nowhere below, between, or above.

Adoption of 273.16 for assignment to the triple-point isotherm was accomplished in 2 steps of which an excellent account is given by Stimson[11]. In 1948 The Ninth General Conference on Weights and Measures[12] had decided that the triple-point isotherm was susceptible of practical realization with greater precision than the ice-point isotherm, had recognized the principle of assigning an exact value of absolute temperature to a single fixed-point isotherm which would be the triple-point isotherm, but had left the particular assignment to be chosen later. The choice was not an easy one to make because it was generally agreed that the new scale to be defined by it should be essentially a refinement of the old scale on which the steam-point and ice-point temperatures were made to differ by exactly 100. Thus the chief difficulty came in deciding what is the current best value of the ice-point temperature on the old scale considering the spread of more than 0.02 in the estimates of the various national laboratories. After long discussion in the Advisory Committee on Thermometry and Calorimetry[13] agreement was reached on 273.15 as best value, subject to the stipulated difference of exactly 100 between steam-point and ice-point temperatures, of the ice-point temperature, whereupon 273.16 was recommended to the International Committee on Weights and Measures[14] and subsequently adopted by the Tenth General Conference[9] as previously

noted. This body being supreme authority in such matters, one is bound to accept $T_1/T_0 = 373.15/273.15$ as a better value of the ratio of steam-point to ice-point temperature than that previously used, namely, $T_1^*/T_0^* = 373.16/273.16$.

Another method of determining the ratio of 2 absolute temperatures, which is of particular interest here, calls for the experimental determination of (1) the quantity of heat $\gamma(p)$ required to maintain pressure constant during withdrawal of unit weight of saturated vapor from an otherwise adiabatic calorimeter of fixed volume containing a saturated mixture of vapor and liquid water, as a function of the constant pressure p; (2) the compressibility factor $Z_g(p)$ of the saturated vapor as a function of pressure p; (3) the specific volume of water vapor at some conveniently accessible point usually taken on the ice-point isotherm having saturation pressure p_0 not necessarily observed. Osborne, Stimson and Ginnings[2,3] have published remarkably accurate determinations of the calorimetric quantity $\gamma(p)$ closely covering the range from ice-point to steam-point pressure; and from the single determination of specific volume in (4), using the compressibility factor in (2), can be computed the zero-pressure value of the product pv for the ice-point isotherm having saturation pressure p_0—call it $A(p_0)$. Thus the quantity,

$$P(p) = -A(p_0)\int_1^p (Z_g/p\gamma)\,dp \qquad \text{.....(1)}$$

is a pure pressure function capable of practical realization with high accuracy; it vanishes, by construction, at a pressure of one standard atmosphere, that is, $P(1) = 0$. The function is important here because, according to thermodynamic theory,

$$T_0/T - T_0/T_1 = P(p), \qquad \text{. . . . . (2)}$$

where

T = temperature on the absolute scale.

T_0 and T_1 = ice-point and steam-point temperatures, respectively.

If p_0 is known, Equation 2 will determine T_1 as soon as a value is assigned to T_0.

Goff and Gratch[1] have used Equation 2 in reverse to calculate saturation pressure p^* as a function of the absolute temperature T^* on the scale with $T^*_0 = 273.16$ and $T^*_1 = 373.16$ after having synthesized the function $P(p)$ from available data. No attempt will be made here to improve the accuracy of $P(p)$ in the light of new experimental findings since any possible improvement is believed to be altogether too slight to justify the effort at this time. No change in $P(p)$ implies that the new and the old scales are related by

$$\frac{T_0}{T} - \frac{T_0}{T_1} = \frac{T_0^*}{T^*} - \frac{T_0^*}{T_1^*} \qquad \text{.....(3)}$$

which it is convenient to rewrite as

$$\frac{t^*-100}{t-100} = \frac{T_0 T_1^* T_1^*}{T_0^* T_1 T_1}\left[1-\left(\frac{T_0^* T_1}{T_0 T_1^*}-1\right)\left(\frac{t^*-100}{T_1^*}\right)\right]$$

with

$$t = T - T_0$$

$$t^* = T^* - T^*_0$$

or, after inserting numerical values and rearranging, as

$$\frac{t-t^*}{0.0016988\left(1-\frac{t^*}{100}\right)}=1 + 0.154754\left(1-\frac{t^*}{100}\right) + \cdots\cdots\cdots\cdots\cdots\cdots(4)$$

Thus, the 1946 Goff-Gratch value p_0^* of the ice-point pressure lies at 273.1519617 on the new Kelvin scale so that the new improved value p_0 at 273.15 must be less

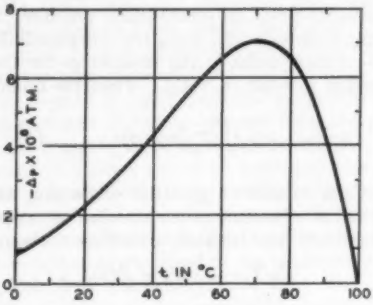

FIG. 1—PLOT OF CORRECTION Δp FROM DATA OF TABLE 1

than p_0^* by amount 0.0019617 $(dp/dT)_0$, approximately. The derivative dp/dT is calculable with requisite accuracy from the Goff-Gratch formulas.

SATURATION WITH RESPECT TO LIQUID, −50 TO 100 C

For saturation with respect to liquid the Goff-Gratch formula is of the form, $\log p^* = G(T_1^*/T^*)$, and therefore yields the improvement

$$\log p = G + \left(\frac{T_1^*}{T^*}\right)^t \left(\frac{(t-t^*)}{T_1^*}\right) G' \cdots\cdots\cdots(5)$$

At $t^* = 0$, for example, compute: $G = -2.2198292$, $G' = -6.310044$, $t - t^* = 0.0019617$, whence $\log p_0 = -2.2198911$ and $p_0 = 0.0060271$ atm as against $p^*_0 = 0.0060280$ atm before improvement. The correction γp to be added to the previously tabulated values of p^* as a function of (thermodynamic) Centigrade temperature t^* to give improved values p as a function of (thermodynamic) Celsius temperature t is exhibited graphically in Fig. 1 and numerically in Table 1.

The next step is to amend the formula to contain these corrections. Writing $G(T^*_1/T^*) = G(T_1/T) + (T^*_1/T^* - T_1/T)G' + \ldots$, calculate the coefficient of G' from Equation 3 to be $(T^*_0T_1/T_0T^*_1 - 1)\ (T^*_1/T^* - 1)$ which is reduced to $(0.01 \times 100/T_0T^*_1)\ (T^*_1/T^* - 1)$ by inserting $T^*_0 - T_0 = 0.01$ and $T_1 - T_0 =$

100. Values of the increment $\Delta G = (T^*_1/T^* - T_1/T)G'$ were computed at 11 equally spaced values of t^* in the range 0(10)100 deg C and adjusted by the method of least squares to the formula,

$$\Delta G = 0.901 \times 10^{-4}\left(\frac{T_1}{T} - 1\right) - 0.811 \times 10^{-4} \log\left(\frac{T_1}{T}\right)$$

with a root mean square deviation of $\pm 1.6 \times 10^{-8}$. The adjusted increment is an easy one to add to $G(T_1/T)$ after which the argument of G is changed to $T_{01}/T = (T_{01}/T_1)\ (T_1/T)$. The final result is

TABLE 1—CORRECTION Δp TO BE ADDED TO 1946 GOFF-GRATCH VALUES OF SATURATION PRESSURE OF WATER WITH RESPECT TO LIQUID

t (°C)	$-\Delta p \times 10^2$ (MB)	p (MB)	p (LB/IN.^{2})	$-\Delta p \times 10^4$ (LB/IN.^{2})	t (°F)
0	0.85	6.1078	0.088586	0.126	32
10	1.39	12.272	.17799	0.207	50
20	2.16	23.373	0.33900	0.322	68
30	3.14	42.430	0.61540	0.467	86
40	4.27	73.777	1.0700	0.636	104
50	5.46	123.40	1.7897	0.813	122
60	6.49	199.26	2.8900	0.967	140
70	7.02	311.69	4.5207	1.045	158
80	6.56	473.67	6.8699	0.977	176
90	4.48	701.13	10.169	0.667	194
100	0.00	1013.25	14.6959	0.000	212

$$\log p = 10.79574\ (1 - T_{01}/T) - 5.02800 \log\ (T/T_{01}) + 1.50475 \times 10^{-4}\ (1 - 10^{-8.2969(T/T_{01}-1)}) + 0.42873 \times 10^{-3}\ (10^{4.76955(1-T_{01}/T)} - 1) - 2.2195768 \quad . \ . \ . \quad (6)$$

with $T_{01} = 273.16$ and with p expressed in standard atmospheres†. The amended Equation 6 gives $p_{01} = 0.00603147$ atm as triple-point pressure and, as previously noted, $p_0 = 0.0060271$ atm as ice-point pressure. The International Meteorological Organization[4] has recommended that the 1946 formula from which the foregoing has been derived be used for undercooled liquid down to −50 C pending further research.

SATURATION WITH RESPECT TO ICE

For saturation with respect to ice the Goff-Gratch formula is of the form, log $p^* = F(T^*_0/T^*)$. The first step is to change the argument of the function F to

†The definition of the standard atmosphere (atm), 1 atm = 1013250 dyne/cm^2, given by the Ninth General Conference on Weights and Measures in defining the International Temperature Scale of 1948 was adopted for all purposes by the Tenth General Conference (see Reference 9).

$T^*_{01}/T^* = (T^*_{01}/T^*)\ (T^*_0/T^*)$ with additive constant $-2.2195136 = \log p^*_{01}$. Then numerical values of the increment, $\Delta F = (T^*_{01}/T^* - T_{01}/T)\ F'$, were computed at 10 equally spaced values of t in the range -180 (20) 0 deg C and adjusted by the method of least squares to the formula,

$$\Delta F = 0.7015 \times 10^{-4} + 0.0288 \times 10^{-4}\left(\frac{T_{sl}}{T} - 1\right) - 0.3419 \times 10^{-4} \log\left(\frac{T_{sl}}{T}\right) + 0.1313 \times 10^{-4}\left(1 - \frac{T}{T_{sl}}\right)$$

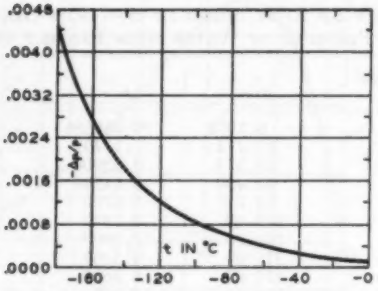

FIG. 2—PLOT OF CORRECTION $\Delta p/p$ FROM DATA OF TABLE 2

with a root mean square deviation of $\pm 2.8 \times 10^{-8}$. This is an easy increment to add to $F(T_{01}/T)$ and the final result, after adding 0.0000069 to make the new saturation curve pass through the new triple-point pressure, is

$$\log p = -9.096853\left(\frac{T_{sl}}{T} - 1\right) - 3.566506 \log\left(\frac{T_{sl}}{T}\right) + 0.876812\left(1 - \frac{T}{T_{sl}}\right) - 2.2195768 \quad \text{(7)}$$

with $T_{01} = 273.16$ and with pressure expressed in standard atmospheres. The amended Equation 7 accepts 0.00603147 atm as triple-point pressure and gives $p_0 = 0.0060265$ atm as ice-point pressure instead of the 0.0060273 atm used by Goff and Gratch[1] in the 1946 formula. The quantity $p^* \log_e(p/p^*)$ calculated from Equation 7 with $T = t + 273.15$ and from the corresponding 1946 formula with $T^* = T + 0.01$ is the correction Δp to be added to the previously tabulated values of p^* as a function of t^* to get improved values of p as a function of (thermo-

dynamic) Celsius temperature t. The corresponding relative correction $\Delta p/p$ is exhibited graphically in Fig. 2 and numerically in Table 2.

SUMMARY

The 1946 Goff-Gratch formulas for the saturation pressure of water, which were adopted by the International Meteorological Organization in 1947 and by the International Joint Committee on Psychrometric Data in 1949, have been amended to give improved values of saturation pressure at stated temperatures on the new Kelvin scale adopted by the Tenth General Conference on Weights and Measures in 1954. The amended formulae are being recommended for adoption to the Commission for Aerology, World Meteorological Organization, at its second session which opens in Paris on June 18, 1957, by the Working Group on Physical

TABLE 2—CORRECTION $\Delta p/p$ TO 1946 GOFF-GRATCH VALUES OF SATURATION PRESSURE OF WATER WITH RESPECT TO ICE

t (°C)	$\Delta p/p$	t (°F)	t (°C)	$\Delta p/p$	t (°F)
−180	−.00469	−292	−80	−.000616	−112
−160	−.00290	−256	−60	−.000431	−76
−140	−.00189	−220	−40	−.000299	−40
−120	−.00128	−184	−20	−.000203	−4
−100	−.000883	−148	− 0	−.000132	+32

Functions and Constants of that Commission. The amended formulas are: Equation 6 for saturation with respect to liquid in the range −50 to 100 C; Equation 7 for saturation with respect to ice below 0.01 C, the triple-point temperature on the new (thermodynamic) Celsius scale. Both formulas give 0.00603147 atm (6.1114 millibars) as triple-point pressure; Equation 6 gives 0.0060271 atm (6.1070 millibars) as ice-point pressure at saturation with respect to liquid; Equation 7 gives 0.0060265 atm (6.1064 millibars) as ice-point pressure at saturation with respect to ice.

Skeleton tables of corrections to be applied to the tabular values of saturation pressure as a function of (thermodynamic) centigrade temperature calculated from the 1946 formulas have been prepared to facilitate revision of these tables to make them give improved values of saturation pressure as a function of (thermodynamic) Celsius temperature instead.

It may not be inappropriate here to note that Reardon[18] has recently calculated the pseudoadiabats of moist air from the Goff-Gratch formulation[5] transposed to temperatures on the new Kelvin scale.

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1608

No. 1608

CRITERIA FOR ROOM NOISE FROM AIR CONDITIONING

By CARLYLE M. ASHLEY*, SYRACUSE, N. Y.

THERE CAN be little doubt about the importance of improved noise control in air-conditioning systems after the surveys of consumer reactions made by DuPont for room air-conditioning units and home air-conditioning systems, and by the *Architectural Forum* for all types of systems. But, many people doubt whether any objective criterion can be set up, which can serve as a standard to distinguish between a satisfactory and an unsatisfactory noise condition. This view has been reinforced by the variation of noise tolerance between individuals and under different surroundings, and the difficulty of simulating the range of auditory response by objective means.

Several criteria have been proposed. These criteria have been based upon the evaluation of different properties of sound, including noise level[1], loudness[2], loudness level[3], speech communication[4], and annoyance[5]. In the 1957 edition of THE GUIDE[6] and in the January 1957 issue of *Noise Control*[7], Dr. Leo L. Beranek has presented 2 related criteria, which he has called *noise criteria* (Figs. 1 and 2). These criteria combine a speech communication criterion and a loudness level criterion. Dr. Beranek cites a substantial amount of background experience in human reactions to noise in support of the *criteria* as a means of predicting acceptable and unacceptable noise situations.

The plan in this paper is to consider the characteristics of noise which make it acceptable or unacceptable, examine the *noise criteria* further, suggest a simplified criterion and a more refined method for the evaluation of *quality* of noise.

CHARACTERISTICS OF NOISE VS. ACCEPTABILITY

The acceptable amount of air-conditioning noise varies greatly, depending upon the use of the space.

Music: A very low noise level can be tolerated where music or speech is to be reproduced, to avoid having the noise obtrude into the dynamic range of the music.

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[1] Exponent numerals refer to References.

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The requirements are only slightly less exacting where music is to be played live, as in auditoriums, churches, music rooms and home living rooms.

Speech Communication: Speech communication is an important requirement of most room uses. But, the background noise level which can be tolerated varies greatly with different applications, depending upon the distance between the speaker and the listener, the acoustical characteristics of the room, and the degree of facility of communication required.

The frequency range which is important to speech intelligibility is from approximately 300 to 5000 cps[8]. The dynamic range between the soft and loud parts of speech is about 30 db, and the variation of the peak level between a soft and very loud voice is approximately 18 db.

When the lower end of the dynamic range of the speech sound drops below the background level, the intelligibility is decreased, and the amount of this decrease depends upon the portion of the dynamic range which is masked out by the background noise. Additionally, speech intelligibility is limited by the sustained reverberation of the speaker's own voice from the preceding syllables. The effect on speech intelligibility is the result of the addition of background noise and the reverberant speech noise.

Calculations of the reverberant speech level have been made for a number of sizes of rooms with the speaker and listener near opposite ends of the room, according to the method of Bolt and McDonald[9].

These computations involve estimating the differential directivity factor of the speech sound and adjusting the acoustical characteristics of the room to provide very good speech communication with normal speech intensity (reverberant level, 25 db below speech level peaks). The levels shown in Fig. 3, are the arithmetic mean of the 3 octave bands between 600 and 4800 cps, which is also the basis used for the *speech communication criterion*.

The large rooms shown on this ideal curve are from practical experience too *dead*. The dashed line gives the values for a room having a maximum recommended sound absorption for speaking. It also assumes that the speaker will slow down his speaking, so as to minimize the effect of the reverberant speech level on intelligibility.

Also shown on this figure is the octave band reverberant speech power level. It is especially significant that the power level remains constant, regardless of the size of the room.

If the background noise including the air-conditioning noise is kept at the same level as that of the reverberant speech sound, the articulation index will be decreased only about 10 percent, and the speech intelligibility will still be very good. Assuming that the noise from other sources can be neglected, the corresponding permissible sound-power level of the air-conditioning equipment will be the reverberant speech power level corrected for the direct component of the air-conditioning noise at the listener position, that is,

$$L'_w = L_w - 10 \log_{10} [(QR/16\pi r^2) + 1] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

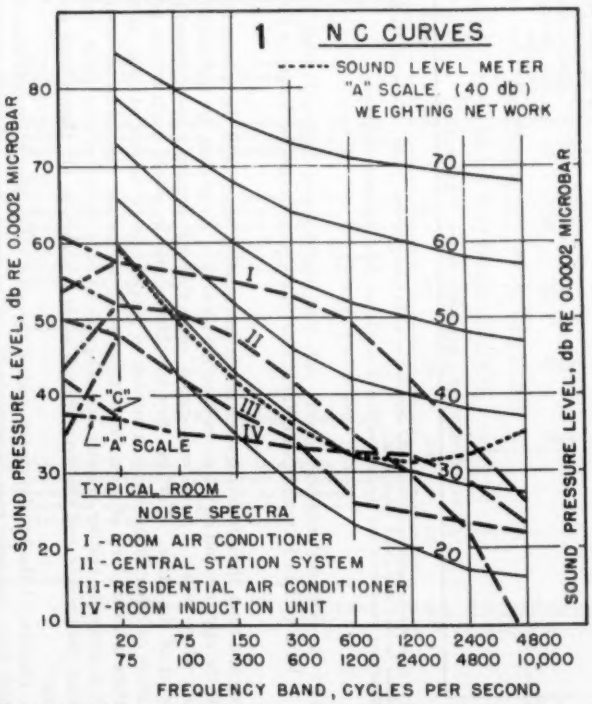
where

L'_w = mean octave band power level from the air-conditioning equipment, 600–4800 cps.

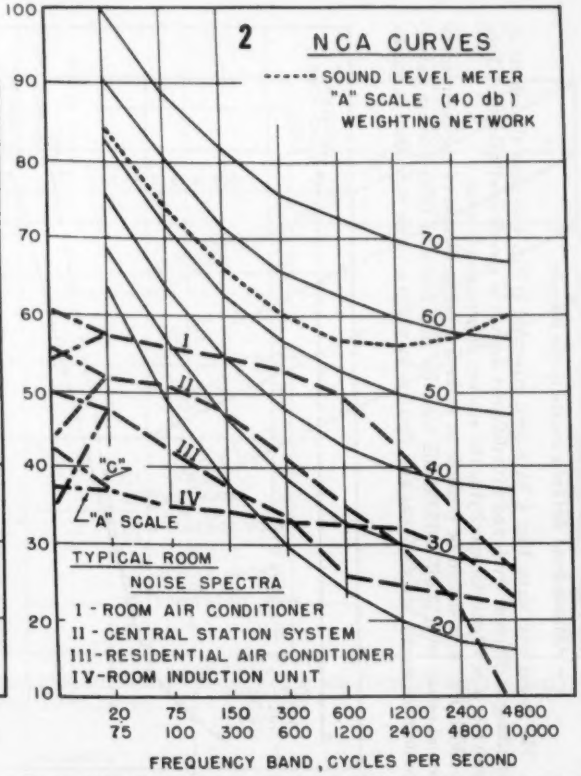
L_w = mean octave band power level equal to the reverberant speech power level for 600 to 4800 cps.

Q = directivity factor.

FIG. 1 (*left*) AND FIG. 2 (*right*)—NOISE CRITERIA CURVES SHOWING TYPICAL ROOM-NOISE SPECTRA



The noise criterion for a given noise spectrum is the highest octave-band value of the spectrum. Example: Noise spectrum No. 1 has a NC value of 45 db, from the peak in the fourth octave band.

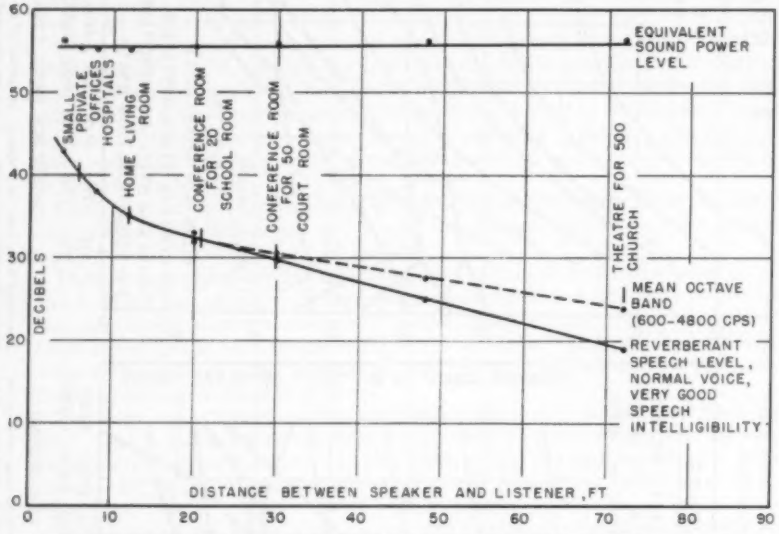


The dotted lines show the frequency weighting of the "A" scale (40 db weighting network of the sound-level meter). These are compared with the NC-30 and the NCA-55 curves.

R = room absorption constant.
r = distance from the noise source to the listener.

The power level data, Fig. 3, have been expanded in Fig. 4 to show the effects of speaking at different levels and with different degrees of speech intelligibility. This provides a new basis for specifying the amount of permissible noise.

With Fig. 4, it should be possible to make a good prediction of the permissible sound-power level of air-conditioning noise for such rooms as offices, living rooms, conference rooms, school rooms and theaters in which a single individual is speak-



The listener is assumed to be close to the opposite end of the room from the speaker. The room has been designed for very good speech intelligibility.
Equivalent sound-power level is calculated from the reverberant speech level using the equation

$$L_w = L_p + 10 \log_{10} \frac{R}{4}$$

FIG. 3—SOUND-PRESSURE LEVEL AND THE EQUIVALENT SOUND-POWER LEVEL DUE TO THE REVERBERANT EFFECT OF SPEECH IN ROOMS OF VARIOUS SIZE

ing at a time, the interfering noise is relatively unimportant, and the listener may be at the opposite end of the room from the speaker. These data are not suitable for use where music is to be reproduced, since this will require a lower sound-power level of the noise. Nor are they suitable for use in stores, restaurants, general offices or other rooms where a number of people may be talking at once, or where there is a high level of noise due to other sources. In this kind of an environment, it is generally neither possible nor desirable to hear speech at any great distance from the speaker, and the direct speech levels as shown by Dr. Beranek are governing[10].

Quality: If the low frequency end of the air-conditioning noise spectrum has a high level, the noise may be annoyingly loud without appreciably affecting speech

The sound-power level is a mean octave-band value for the 600 to 4800 cps bands.

This figure can be used for determining the permissible air-conditioning noise level by using Equation 1 as illustrated in following example:

Example: Assume that it is desired to have fair speech intelligibility with a normal voice in a room having a room absorption constant of 400 and with the directivity factor of the noise of 4 and at a distance from the noise source of 6 ft, find the permissible mean octave band power level of the air-conditioning equipment.

From Fig. 4,

$$QR/16\pi r^2 = 0.9 \quad \ldots\ldots\ldots \quad (1)$$

$$L_w = 62$$

$$L'_w = 62 - 10 \log_{10} 1.9 = 59 \text{ db}$$

This figure is usable only where a single person will be speaking at a distance from the listener in a room without amplification. It also assumes that the sound power in the low frequency bands is not controlling of the design requirements.

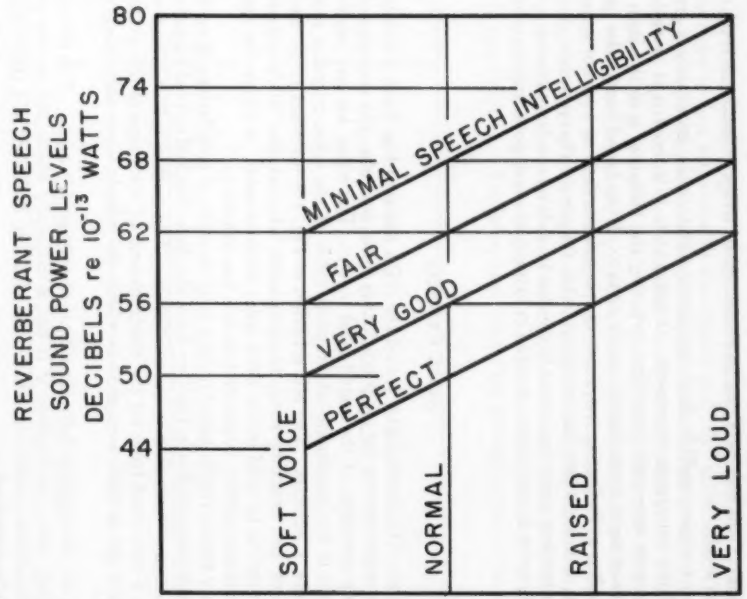


FIG. 4—RELATIONSHIP OF REVERBERANT SPEECH SOUND-POWER LEVEL TO STRENGTH OF VOICE AND SPEECH INTELLIGIBILITY (AN EXTENSION OF THE DATA OF FIG. 3)

communication. This loudness can be evaluated in terms of loudness level or in terms of noise level.

If the noise contains audible single-frequency components, its *quality* is greatly reduced, and the annoyance increased. When a noise is intermittent or varies in loudness, it can be extremely annoying—whether the variation is in the form of a beat, a turbulent noise, a mechanical rattle or tap. Even broad band noise can be annoying if it has a distinctive frequency characteristic, such as a hiss or a roar.

Conversely, a noise which has no identifiable frequency characteristic or time variation is much less annoying, and a higher loudness level can be tolerated. Such a good *quality* noise is sometimes of value in helping to mask poor quality environmental noise.

NOISE CRITERIA

Plotted on the NC curve of Fig. 1, and the NCA curve of Fig. 2 are a number of rather typical noise spectra for air-conditioning noise, and also, the response of the "A" weighting networks of the sound-level meter. The value for any particular noise spectrum is the maximum octave-band value of the noise criterion reached by a given noise spectrum. The NC criterion of Fig. 1 is intended to be used for relatively quiet and exacting room requirements, whereas the NCA criterion of Fig. 2 is intended to be used in the more noisy environments, or where the quality and speech communication requirements are less exacting.

Not only are these criteria based upon a large number of correlations of acceptability with noise characteristics, but they have also been checked out for a number of trouble jobs, under controlled noise conditions. The author believes that these criteria form a very good basis for the establishment of standards of acceptability. However, there are at least 3 respects in which the criteria leave something to be desired.

First, they do not evaluate fully the effects of poor *quality* noise. The effect of a single-frequency noise, particularly if it has a frequency above 200 to 300 cycles per second, is not adequately identified. In addition, the effect of a time variable noise is not fully identified.

The second problem is which of the 2 charts to select for a given problem and the possible need for interpolation between them. It would be better if it were possible to present a single criterion, although in any case, one must recognize the need for some range in the choice of job quality.

The third possible objection to these criteria is the need for analyzing on the basis of octave bands. The author is fully in agreement with the importance of the use of the octave-band analyzer as an analytical and design tool, but it seems obvious that noise standards would be more readily acceptable if they required less measuring equipment, and simpler measuring techniques.

Noise Level as a Criterion: On Fig. 1, the shape of the equivalent "A" scale or 40 db network response of the sound-level meter is plotted at the NC-30 line, and on Fig. 2 at the NCA-55 line. It is significant that the low-frequency portion of the "A" scale reading corresponds very closely to the shape of the NC and the NCA curves, respectively. At the high frequency end, the air-conditioning noise is normally not significant. Therefore, the deviation from the curve at this point is usually of no great importance. It is also of significance that the levels at which the "A" scale reading and the *criteria* curves correspond are approximately those levels at which the 2 *criteria* are most frequently recommended for use.

It is evident from Figs. 1 and 2 that air-conditioning noise typically tends to follow more or less the shape of the *criteria* and "A" scale lines. If the octave-band spectrum of the noise followed the noise-criteria curve very closely, it would be expected that the "A" scale reading would be approximately 10 db higher than the corresponding criterion because of the addition of noise from the 8 octave bands. On the other hand, if the noise spectrum followed the noise-criterion curve very poorly, the "A" scale reading would still be expected to be 5 db higher than the noise criterion due to the effect of the addition of the various octave bands.

A number of different types of sound spectra have been compared as to their noise level and criterion value using the shape of the criterion corresponding to NC-30 and NCA-55. The computed results are plotted in Fig. 5 *vs.* the difference

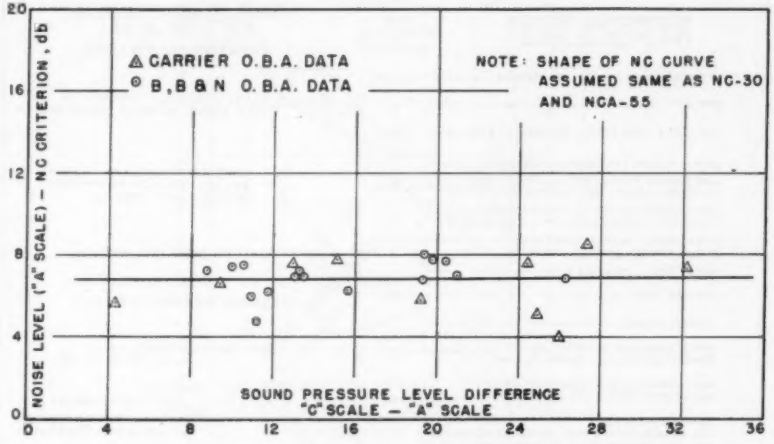


FIG. 5—COMPARISON OF NOISE LEVEL AND NC NOISE CRITERIA FOR A NUMBER OF DIFFERENT NOISE SPECTRA

between the "A" and the "C" scale of the sound level meter. It is evident that the range of the noise level values is roughly from 5 to 9 db above the value of the NC type criteria.

The principal objection to use of noise level as a criterion is that it does not distinguish as completely between poorly shaped curves as the *noise criteria*. However, the magnitude of this difference is, at the most, of the order of 4 db. Furthermore, there is a possibility of using the difference between the "A" and the "C" scale of the sound-level meter for further interpretation of the loudness level effect, if desired.

The record of corrected trouble jobs reported by Dr. Beranek was also reviewed. The value of the corrections can be as readily identified by means of the noise-level readings as by the *noise criteria* evaluation. Again, the differences between the noise level and *noise criteria* are in the range 5 to 9 db.

It should be strongly emphasized that the use of noise level using the sound-level meter is intended only for judgment of the acceptability of the noise in a space.

For the purposes of noise-rating equipment, for proper system design and for the analysis of the results, the use of sound-power level or sound-pressure level on an octave-band basis is imperative.

RECOMMENDED ROOM-NOISE LEVELS AND TYPICAL NOISE LEVELS FROM AIR-CONDITIONING EQUIPMENT

Fig. 6 shows the recommended noise levels for rooms based on the use of the "A" scale noise level as a criterion. While these values are generally consistent

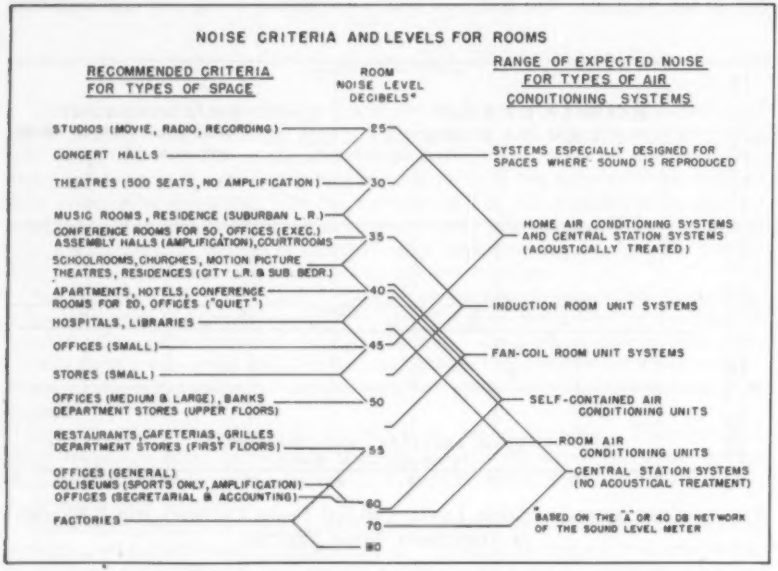

NOISE CRITERIA AND LEVELS FOR ROOMS
RECOMMENDED CRITERIA FOR TYPES OF SPACE
ROOM NOISE LEVEL DECIBELS*
RANGE OF EXPECTED NOISE FOR TYPES OF AIR CONDITIONING SYSTEMS
STUDIOS (MOVIE, RADIO, RECORDING)
CONCERT HALLS
THEATRES (500 SEATS, NO AMPLIFICATION)
MUSIC ROOMS, RESIDENCE (SUBURBAN L.R.)
CONFERENCE ROOMS FOR 50, OFFICES (EXEC.)
ASSEMBLY HALLS (AMPLIFICATION), COURTROOMS
SCHOOLROOMS, CHURCHES, MOTION PICTURE THEATRES, RESIDENCES (CITY L.R. & SUB. BEDR.)
APARTMENTS, HOTELS, CONFERENCE ROOMS FOR 20, OFFICES ("QUIET")
HOSPITALS, LIBRARIES
OFFICES (SMALL)
STORES (SMALL)
OFFICES (MEDIUM & LARGE), BANKS
DEPARTMENT STORES (UPPER FLOORS)
RESTAURANTS, CAFETERIAS, GRILLES
DEPARTMENT STORES (FIRST FLOORS)
OFFICES (GENERAL)
COLISEUMS (SPORTS ONLY, AMPLIFICATION)
OFFICES (SECRETARIAL & ACCOUNTING)
FACTORIES
25
30
35
40
45
50
55
60
70
80
SYSTEMS ESPECIALLY DESIGNED FOR SPACES WHERE SOUND IS REPRODUCED
HOME AIR CONDITIONING SYSTEMS AND CENTRAL STATION SYSTEMS (ACOUSTICALLY TREATED)
INDUCTION ROOM UNIT SYSTEMS
FAN-COIL ROOM UNIT SYSTEMS
SELF-CONTAINED AIR CONDITIONING UNITS
ROOM AIR CONDITIONING UNITS
CENTRAL STATION SYSTEMS (NO ACOUSTICAL TREATMENT)
* BASED ON THE "A" OR 40 DB NETWORK OF THE SOUND LEVEL METER

FIG. 6—NOISE CRITERIA AND LEVELS FOR ROOMS

with the values shown in THE GUIDE 1957 of ASHAE, they have, in some cases, been adjusted according to the information shown in Fig. 3. As has been previously indicated, it is necessary to add 5 to 9 db to the NC values to obtain the equivalent noise-level values. This same addition can also be made to the values of the curves in Figs. 3 and 4.

The values shown are intended to be reasonably representative of good practice. It cannot be emphasized too strongly that judgment must be exercised in assigning a value for each specific situation. In general, the noise levels shown are lower than most present day practice, particularly where using unitary type of equipment.

Extensive experience in homes indicates a noise level of 40 db as a desirable maximum, with 30 to 35 db preferred for quiet neighborhoods. Likewise, the desirable noise level for hotels and apartments is 40 db, except in very noisy environments. There is, also a wealth of experience in office applications, which is

embodied in the material presented. There is relatively less experience for such applications as hospitals, court rooms, libraries, school rooms and the like, and it is the author's belief that substantially more field experience should be gathered and the recommended values modified accordingly.

On the right-hand side of Fig. 6 are shown representative room-noise levels resulting from various types of air-conditioning equipment. It can be seen that there are types of system or equipment available to handle any room use. But, much of the equipment now in use is not fully acceptable for the noise requirements of the

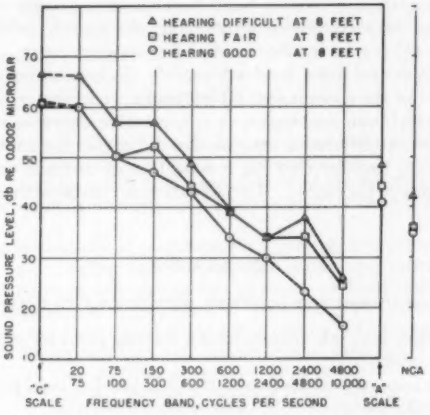


FIG. 7—SPECTRA OF NOISE, NOISE LEVEL AND NCA CRITERIA TAKEN IN A SMALL CONFERENCE ROOM

spaces in which they are being used. This is particularly true of room air-conditioning and self-contained units.

EVALUATING QUALITY

None of the criteria so far proposed is fully satisfactory in the evaluation of the *quality* of the noise as distinguished from the speech interference and the loudness. It should theoretically be possible to evaluate single-frequency components by measurement with a narrow-band analyzer. However, it is not believed that such methods are practical to apply in the field at the present time. It is suggested that consideration be given to applying an additive factor of the order of 5 to 10 db where a single-frequency component is noticeable, the magnitude of the factor to be related to the strength of the single-frequency component.

With respect to judging the effect of variations in noise level, it is believed that a practical, objective tool is available in the form of the *fast* scale of the sound level meter. Maximum and minimum readings should be taken with both the "A" and the "C" scale, and these should then be correlated with subjective observations

of noise quality to determine the proper use of the meter for best evaluation. It is believed possible that the maximum reading of the needle may be a good index.

Another approach to the problem of quality is to assume that any job which is noticeably deficient in *quality* needs to have corrective measures. In such cases, the quality criterion would be used, not for quantitative evaluation, but, for purposes of judging the acceptability without respect to noise level. It is evident that the question of *quality* should have further consideration and study.

FIELD OBSERVATIONS

Fig. 7 shows correlation of octave-band measurements, noise criteria and noise-level measurements with comments concerning the speech intelligibility. These tests were run in a 14½- x 16-ft office used as a conference room. It is evident that both the *noise criteria* and noise level adequately distinguish between the different degrees of the effect of the room noise. The readings emphasize the relatively small increase in level which can accompany a considerable decrease in speech intelligibility. This important difference may be due in part to the sound source being at one side of the room, thereby creating a condition of unequal noise-level distribution in different parts of the room. This situation is typical of many air-conditioned rooms.

CONCLUSIONS

The following conclusions can be drawn with respect to noise criteria:

1. It is believed that available criteria form a starting point for establishing standards of acceptable room noise.

2. *Noise level* has been re-examined as a criterion, and has been found to have several advantages and good correlation with other accepted criteria.

3. Further information has been presented with respect to the correlation of speech communication with distance in typical rooms.

4. A new method has been presented for determining the sound-power level requirements of the air-conditioning noise of certain types of rooms directly, without the necessity of going through sound-pressure level.

5. The need for further study has been pointed out, particularly with respect to evaluating the effect of quality of noise having bad characteristics, and for obtaining more field experience on acceptable noise levels for certain types of room uses.

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2. Calculation of the Loudness of Complex Noise, by S. S. Stevens (*Journal of the Acoustical Society of America*, Vol. 28, pp. 807–832, 1956). (This supersedes earlier methods of calculating loudness).

3. The Loudness of Bands of Noise, by Irwin Pollack (*Journal of the Acoustical Society of America*, Vol. 24, 1952, pp. 533–538). *Note*: Portions of these data have previously been presented by Beranek, Peterson, Marshall and Cudworth in the same *Journal*, Vol. 23, 1951, p. 261. (Loudness level of a broad band sound is now computed from loudness. See Ref. 2).

4. *Acoustics*, by L. L. Beranek (McGraw-Hill Book Co., Inc., New York, 1954, pp. 419–429).

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6. HEATING VENTILATING AIR CONDITIONING GUIDE 1957, Chapter 40—Sound Control (published by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, New York, N. Y.).

7. Criteria for Noise in Buildings, by L. L. Beranek (*Noise Control*, January 1957, pp. 19–27).

8. *Acoustics*, by L. L. Beranek (McGraw-Hill Book Co., Inc., New York, 1954, pp. 406–417).

9. Theory of Speech Masking by Reverberation, by R. H. Bolt and A. D. McDonald (*Journal of the Acoustical Soicety of America*, Vol. 2, pp. 577–580, 1949). (See also Reference 8).

10. *Acoustics*, by L. L. Beranek (McGraw-Hill Book Co., Inc., New York, 1954, Table 13.8, p. 420).

DISCUSSION

R. A. GONZALEZ, Dayton, Ohio: Mr. Ashley has given some excellent and practical guidance on sound-level criteria as applied to air-conditioned spaces. This guidance helps to define the objective, which is the sound level of the finished room . . . and this sound level is not determined entirely by the sound level of the air-conditioning equipment.

The sound level of the completed installation is a composite of many factors including details of surroundings, building construction, decoration and furnishings. Also, the details of equipment location and installation methods are most important to the final result. It is my suggestion, as an extention of Mr. Ashley's excellent paper, that continual emphasis needs to be placed on these other factors. It appears particularly important to secure a full appreciation and recognition of these facts by architects, engineers and installers, as they can frequently exercise greater control on the final acoustical result than can be exercised by the equipment manufacturer.

JOHN EVERETTS, JR., Philadelphia, Pa.: Curve II, Fig. No. 1 and No. 2 indicate the noise level for central station systems. Would this vary greatly whether higher speed centrifugal machines or lower speed reciprocating machines were used?

J. B. CHADDOCK†, Cambridge, Mass.: The use of speech levels as a criterion for noise control has been suggested by a number of investigators of noise criteria. Dr. Leo L. Beranek, whose work has been referred to in Mr. Ashley's paper, based his original criterion curves on ease of speech communication in rooms. However, later studies by Beranek indicated that a criterion based on speech levels alone was not satisfactory. Communication by speech may be entirely satisfactory within an office space, yet occupants may complain of annoying background noise when there is a relatively high level of low-frequency sound present. I wonder if Mr. Ashley would comment on this aspect of annoying low-frequency sound when using reverberant-speech level as a noise criterion.

Another item on which I should like to comment is the use of the A-scale of the sound level meter. I believe that Mr. Ashley would agree that for the purpose of intercomparing equipment whose sound-frequency spectrums are similar, and that do not have a large portion of their noise concentrated in the low-frequency bands, A-scale readings are useful. However, for intercomparing old and new equipment, or equipment with significantly different sources of noise generation, an octave-band analysis should be used. The reason being that whether a piece of equipment is annoying or not may depend on the relative amounts of low and high-frequency noise it generates, and this cannot be determined with A-scale readings.

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† Massachusetts Institute of Technology.

LESTER T. AVERY, Cleveland, Ohio: I am under the impression that air-conditioning units show a different noise level depending on their location. It would be interesting if Mr. Ashley could furnish further information concerning this impression of mine.

H. E. KINNEY, Detroit, Mich.: Since both Mr. Ashley and myself wear hearing aids, it might be worthwhile to point out, hearing aid consultants tell us, that consonants in our speech have more cycles per second than vowels.

Also, I am told by ear doctors that most people who have a hearing loss have a greater loss on the higher frequencies than on the low. This being the case, partially deaf people would not be disturbed by higher noise levels as much as normal hearing people, particularly in the higher cps.

AUTHOR'S CLOSURE: In reply to Mr. Kinney's question, I would say that people with impaired hearing who wear suitable hearing aids should be less affected by low-frequency noise than people with normal hearing because most hearing aids either do not respond well to sounds below 500 cycles per second or have a means of attenuating these low-frequency sounds, thus, concentrating the hearing in the higher-frequency range, which is important to speech.

Mr. Avery is certainly correct that air-conditioning units produce a different room-noise level under different conditions of application. Where the unit is located entirely in the room, the characteristics of the room and the location of the observer in relation to the unit greatly influence the noise level. Where the unit is located outside the room, the attenuation of the transmission through the ducts and through the walls of the room also influence the noise level in the room.

In response to Mr. Everetts' question, Curve II in Figs. 1 and 2 were intended to represent the noise from the air-conditioning system of a central station system, rather than that from the refrigeration machine, which should be selected and applied in such a manner as to produce a negligible noise in the occupied area. Centrifugal refrigerating machines are usually easier to apply so as to avoid the transmission of noise to air-conditioned spaces than are reciprocating machines, particularly on the upper floors of buildings.

The comments of Mr. Gonzalez are very pertinent. The room noise is a function of the sound treatment of the room and the location of the equipment or outlets, as well as the sound-power level. Also, in many cases, difficulties have arisen due to inadequate isolation of the equipment in the apparatus room or to the use of thin ceilings under duct plenums. Another frequent source of noise to the room is through an opening cut in the door to provide a return for the air-conditioning air.

In response to Dr. Chaddock's discussion, I would like to comment that one of the special values of the use of the "A" scale is that it does take into account the possibility of annoying background noise of very low frequency, which was not properly evaluated by Dr. Beranek's earlier criterion of *speech communication*. The main criticism of it on this score is that it is not as flexible with respect to very low or very high levels as the NC or NCA curves, which are based upon octave-band analysis.

While it is probably true that the use of the "A" scale of the sound-level meter as a criterion is best adapted to compare sounds of similar frequency spectrum, I believe that the "A" scale is useable even though the spectra may be very different. I have already shown that the "A" scale criterion is useable where very low-frequency sound is an important part of the spectrum. For sounds of predominantly higher frequency, Dr. Beranek's criteria correspond very closely with his earlier *speech communication* criterion. For this situation, it can be shown that the "A" scale reading will vary from approximately 5 and 9 db above the speech communication criterion, depending upon the frequency distribution of sound. This is the same range which I have shown when including lower-frequency sounds. The principal defect of the "A" scale reading is that the weighing network does not continue to slope off at the frequencies above 1,000 cycles. It would be very desirable to have a new network, which had this characteristic.

1609

No. 1609

EVALUATION OF FOUR METHODS FOR DETERMINING SOUND-POWER OUTPUT OF A FAN

By W. F. KERKA*, CLEVELAND, OHIO

This paper is the result of research carried on by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio.

IN THE PAST, several methods have been employed for measuring equipment noise. In general, the methods have been based upon overall noise levels only, with little regard to the frequency distribution. In any noise-reduction problem, a knowledge of the frequency distribution, at least, by octave bands, is necessary for effective control.[1] It is therefore the purpose of this investigation to evaluate 4 methods of determining acoustic-power output by octave-band analysis. The program was conducted under the guidance of the Technical Advisory Committee† on Sound and Vibration Control.

A centrifugal fan was chosen as a noise source since it represents a common component of air-conditioning systems. Measurements of acoustic power were made in a reverberant space, in a free-field, within the duct using an anechoic termination at the end, and at the duct opening. The test results have been expressed as watts of acoustic power in each octave band as this is an absolute unit of measurement. For ease of comparison the results have also been expressed on a power-level basis in decibels. Measurements of inlet and discharge noise were made at fan speeds of 1110 and 1340 rpm and at 2 system resistances for each speed. The test results for only one of these 4 operating conditions are shown in the paper. It is the purpose of this report to compare the test methods only and not to correlate the acoustic-power output for different operating conditions of the fan.

TEST APPARATUS

Fan and Duct Systems: The fan installation is shown in Fig. 1, and the 15-in. test duct extending 10 ft into the reverberant room is shown in Fig. 2. The cen-

* Research Engineer, ASHAE Research Laboratory. Member of ASHAE.

† Personnel: H. A. Lockhart, *chairman*; A. F. Hubbard, *vice chairman*; C. H. Allen, Sr., C. M. Ashley, A. E. DeSomma, Sidney Gordon, R. N. Hamme, H. C. Hardy, E. M. Herrmann, C. R. Hiers, F. B. Holgate, D. H. Krans, R. D. Lemmerman, R. D. Madison, R. E. Parker, T. A. Walters, R. J. Wells.

[1] Exponent numerals refer to References.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Murray Bay, Que., Canada, June 1957.

trifugal-type fan had 36 forward-curved blades and a 15 in. wheel diameter. A 3-phase, 5-hp motor drove the fan through a variable-pitch pulley. The duct system was designed with one end located in the upper Laboratory and the other terminating in the reverberant test room below. The 2 openings were sufficiently separated so that the noise radiating from one would not interfere with measurements made at the other. Banks of glass-fiber filters located at the end of the duct in the upper Laboratory were used to load the system by changing the depth and face area of the filters. A 20-station pitot tube traverse was employed for de-

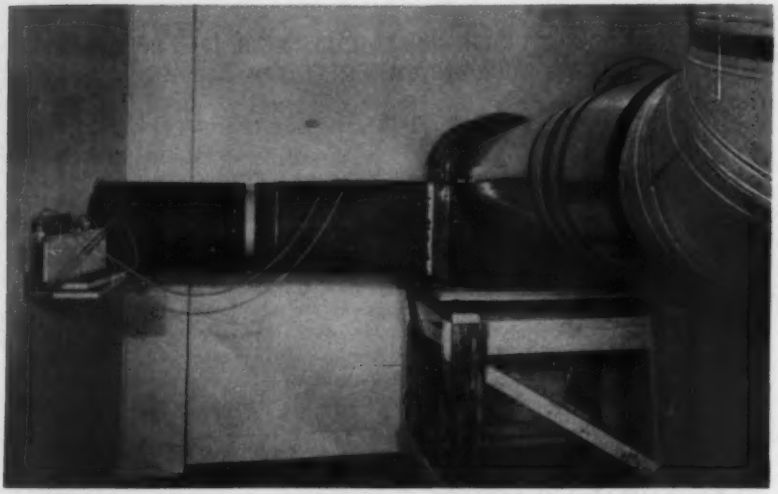

FIG. 1—FAN ROOM INSTALLATION

termining air-flow quantities. The fan and motor were mounted on a sheet of 1½ in. thick plywood which was supported by 4 vibration isolators attached to a wooden base.

The duct was coated with ¼ in. of automotive undercoating to reduce resonance. Measurements of inlet or discharge noise could be made by having the fan exhaust air from the room, or by reversing the fan connections and allowing it to discharge into the room. The fan was isolated from the duct system by canvas couplings. The fan room, which measured 14½ x 9 x 10½ ft high, was separated from the reverberant room by a sound-barrier wall. A separate system of studs was used for each side of the wall, and 2 thicknesses of gypsum board were nailed on each surface. A refrigerator-type door with double seals was mounted in the wall.

The Reverberant Room: The reverberant room measured 30 x 26½ x 11 ft high. The volume was about 8900 cu ft and the surface area about 3050 sq ft. The floor was terrazzo and the walls and ceiling were surfaced with rough hard plaster. A 4-ft high tile wainscot extended around the room. Two 1 x 2 ft plastered beams

ran across the ceiling. The surfaces of the fan room were identical to the test room. Contained in the test room were pieces of equipment and a number of plaster panels one of which is shown in Fig. 2. The presence of the equipment and stored panels helped to create a diffuse sound field. All of the doors in the reverberant room were closed during tests, except one which allowed the air to circulate through a stairwell and several other rooms until it reached the upper Laboratory where the other end of the duct system was located. This arrangement provided a recirculating system. The sound-measuring apparatus was located on a long table also shown in the figure. The operator remained at the table when acoustic measurements were made and the location of the stored equipment was not changed.

FIG. 2—THE REVERBERANT TEST ROOM

Free-Field Setup: For the free-field tests, the fan and the duct system were moved to a rear room of the Laboratory. In this case the 15-in. test duct extended 10 ft through the rear wall into the outdoor space and about 9 ft above the ground level, as shown in Fig. 3. The *free-field*, was clear of obstructions except for the ground and a brickwall 35 ft to one side of the test duct. Tests were conducted during the summer when the weather was clear and the wind was at a minimum.

Anechoic Termination: For measurements made within the test duct, an anechoic termination, exponential horn, and a round-to-square transition section were coupled to the duct. The termination was similar to the one described by Beranek *et al.*[2] The shell of the termination was made from ⅜-in. plywood, while the horn was fabricated from 20-gauge sheet metal, and covered with a ¼-in. thick layer of undercoating. The termination was lined with 1-in. thick glass fiber, and 3 glass fiber sound-absorbing wedges (each 7¼ x 22 x 22 in. long) were mounted together (with the apexes pointed toward the noise source) and suspended at the center leaving a 2-in. periphery for air flow.

The Standard Source: A standard noise source used for calibrating a reverberant space must have sufficient sound power in each octave band to be well above the background level. Since the background level (both indoors and outdoors) is usually predominant in the lower frequencies, the standard source must be designed

to give a measurable amount of low-frequency sound. The source used, shown in Fig. 4, met the needed requirement. The unit consisted of an industrial exhauster with a modified housing. The fan wheel contained 5 blades and was 18 in. in diameter. The inside width of the housing was 9½ in. and the periphery was open except for the 6 stator blades shown in the figure. The inlet was 7½ in. in diameter. The modified exhauster gave sufficient sound power in all octave bands,

FIG. 3—FREE-FIELD TEST SETUP

and no corrections for background noise were necessary. Corrections are necessary when the background level is less than 10 db below the level being measured.

Instrumentation: Acoustic measurements were made with a General Radio, Type 1551-A, sound-level meter coupled to a Scott, type 420-A frequency analyzer used for octave-band analysis. The microphone was of the standard Rochelle-salt crystal type supplied with the sound-level meter. A 25 ft. cable was used between the microphone and the meter. Because a crystal microphone is affected by temperature and the insertion of a cable, corrections to the readings of the sound-level meter were applied. At the higher frequencies the response of the microphone is dependent upon the incidence of direct sound. The flattest response occurs for random incidence, or when the microphone is located in a completely diffused field. To obtain this *random response* for measurements of direct sound, the micro-

phone was oriented so that is axis was between 55 and 75 deg with the line joining the source and the microphone. For measurements made within the duct where a 90 deg incidence angle occurred, a correction was applied in the last 3 octave bands to obtain the random response.

The reverberation time of the test room and the standing-wave patterns in the test duct were determined with a Bruel and Kjaer one-third octave-band analyzer, recorder, and condenser microphone. The Rochelle-salt microphone was cali-

FIG. 4—THE STANDARD SOURCE

brated against the condenser microphone by taking a series of duplicate readings. The one-third octave-band readings (corrected for the microphone calibration) were added logarithmically to give octave-band readings and these were compared with the readings from other instruments. Agreement was within +1.5 db. to −1.0 db. The inherent error of sound-measuring apparatus is about ±1 db. The sound-level meter was checked daily using the General Radio sound-level calibrator.

TEST PROCEDURE

Reverberant-Room Method: In a reverberant field, the sound-pressure level of a noise source is dependent primarily upon the physical dimensions of the room and the sound-absorbing qualities of its surfaces. The room constants for each octave band take these 2 factors into account. The room constants can be found by several methods. In one method, the reverberation time for each octave band must be determined, and in another method the sound-pressure level in the reverberant field of a standard-noise source of a known acoustic output must be measured. Both techniques were employed to calibrate the room.

In the first method Equation 1[3] was used when the reverberation time for each octave had been determined by measuring the rate of decay of an impact noise.

$$R = (0.05V/T)\,[1 + (0.025V/TS)] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where

R = room constant, square feet.
V = volume, cubic feet.
S = surface area, square feet.
T = reverberation time, seconds.

In the second method, the acoustic power of the standard source must be known before the room can be calibrated. The acoustic power was determined by placing the source on a concrete apron in an outdoor space, and by integrating the incremental sound power over a hemispherical surface having a radius of 10 ft. The source was then placed on the floor of the reverberant room beneath the test duct opening. Octave-band readings were taken at 8 positions in the room at a minimum distance of 8 ft from the source and as far from any reflecting surface as space would permit. The microphone was always located midway between the floor and bottom of the ceiling beams. The maximum deviation of most of the readings from the mean in each octave for the 8 positions was ±1 db. Several readings in the second and third bands had a maximum deviation of ±2.5 db. A mean of the readings in each octave band was used in Equation 2[4] to determine the room constants.

$$R = W/\{\text{antilog}_{10}\,[(SPL_R/10) - 13.65]\} \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where

W = acoustic power of the source (watts).
SPL_R = sound-pressure level in the reverberant field (decibels).

The acoustic power of the sound radiating from the inlet or discharge ducts of the fan into the reverberant space was then determined by again using Equation 2. In this case, W, the acoustic power, was the unknown. Instead of taking sound-level readings at 8 positions in the room for each operating condition of the fan studied, 3 positions were chosen where the deviations from the mean of the 8 positions were within ±1 db.

Equation 2 as applied to the standard source can be combined with Equation 2 as applied to the unknown source with the resulting relationship identical to the one used by Bishop and Hardy[5]. The room constant is thus eliminated from the calculation.

Measurements within the Test Duct: To measure the acoustic power delivered by the fan to the inlet and discharge ducts of the fan, 2 difficulties are encountered. First, the pressure wave in the lower frequencies is reflected back toward the source from the open end.[1] To eliminate this, the pressure wave generated by the source must be completely absorbed by a termination. Practically, this is difficult to achieve since the effectiveness of the termination is usually limited by the cutoff frequency of the coupling (horn) between the duct and the termination. The horn should therefore be designed with as low a cutoff frequency as is feasible. The second consideration is the self-generated noise that occurs whenever a microphone is placed in a moving air stream. To reduce this level sufficiently below the noise level being measured, a windscreen must be placed over the microphone. The wind-screen used in this study was identical to the one described by Beranek.[2]

The screen was in the shape of a cylinder, 3-in. dia. x 5-in. long, and made of ¼-in. mesh hardware screen covered with parachute nylon having about 80 threads to 1 in.

The microphone and windscreen were located 12 in. from the end of the test duct. Readings were taken at the duct centerline and 2 in. from the centerline. Since the windscreen decreased the sensitivity of the microphone in the last 2 octave bands, a correction was applied. In addition, an orientation correction[4] was made in the last 3 octave bands since the centerline of the microphone was placed at right angles to the centerline of the duct. The acoustic power in the duct was determined from Equation 3, the derivation of which is shown in the Appendix. Use of this equation is restricted to the assumption that the sound waves within the duct are plane waves. This is approximately the case[1].

$$W = 10^{-13} S \,[\text{antilog}_{10}\,(SPL/10)] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where

S = cross-sectional area of the duct, square feet.
SPL = sound pressure level within the duct, decibels.

Free-Field Test: The acoustic power radiated from the open-end duct into a free space was determined by taking measurements at the center of 10 equal areas on each of 2 spherical surfaces shown in Fig. 5, one of 4-ft radius and one of 8-ft radius with their centers located at the duct opening. The following relationship was used.

$$W = 10^{-13} \Sigma \Delta S \,[\text{antilog}_{10}\,(SPL_s/10)] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

where

ΔS = area of each increment, square feet.
SPL_s = sound-pressure level at the center of each area, decibels.

The total acoustic power was found by adding the sound power in each incremental area. During all readings the windscreen was placed over the microphone, mainly to reduce the heat gain from the sun. Ambient air temperatures were constantly taken to determine the necessary corrections to the microphone readings. Although a difference in temperature between the microphone and the point of measurement could have existed, an error of 10 F deg would have produced an error in reading of less than 1.5 db.

During sound-level measurements at the discharge duct, it was necessary to make corrections to readings in the lower octave bands, for the one position along the duct centerline, in the wind stream. The readings at this position were high, and this was apparently due to the turbulent noise generated in the wind stream when acting as a jet[6]. This turbulent noise is not radiated from the duct opening or caused by the air stream impinging on the microphone or windscreen. Because it exists in the stream with positive and negative pressure points much closer than a wave length, this turbulent noise is not radiated to the surroundings[6]. The readings were corrected by taking octave-band measurements at 1, 2, 4, and 8 ft along the centerline to observe the 6 db drop that normally occurs with doubling of distance. At the 600–1200 cps band and above this was apparent while in the lower frequencies the readings generally increased or did not drop off as rapidly as theory would predict. In the first 4 bands at both 4 and 8 ft, the centerline readings were corrected by reducing them to the levels that would be expected with the 6 db drop-per-doubling effect. Although the corrected readings compared with

readings taken just outside the airstream, it is recommended that this position be avoided. No corrections were necessary during the inlet studies since all measurements were made at points beyond the boundries of the airstream. A plot of the directivity pattern for both inlet and discharge flow gave a fairly uniform pattern, with the readings in the higher frequencies at positions 6 through 10, somewhat lower than at positions 1 through 5 of Fig. 5.

Duct-Opening Test: The duct-opening test is unique in that it does not require an elaborate test setup. Since measurements are made at the outlet, the exact characteristics of the room in which the test duct is located need not be known. Under no-flow conditions, sound-level readings were taken at 2 positions near the

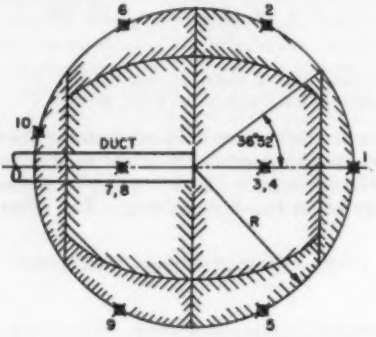

FIG. 5—MICROPHONE POSITIONS USED FOR FREE-FIELD TEST METHOD—POSITIONS ARE AT CENTER OF INCREMENTAL AREAS

periphery of the duct opening (other end closed off) and at 2 positions on the face of the duct. The mean values of the octave-band readings taken at the face (each position of measurement being midway between the center of the duct opening and the periphery) were substituted in Equation 3 to determine the acoustic power at the duct opening for no flow.

Under flow conditions, readings were again taken at the periphery of the duct opening. For discharge studies, the edge of the microphone was placed about ¼ in. from the peripheral edge of the 1½-in. angle around the opening as this was beyond the boundaries of the air stream. For inlet studies, however, the air motion at this location on the periphery was noticeable, hence the microphone was located 4 in. from the angle (for both flow and no-flow conditions) as shown in Fig. 2. The peripheral readings were taken on opposite sides of the opening.

The values obtained from the preceding measurements were substituted into Equation 5 to determine the acoustic power at the duct opening with flow. A derivation of this equation is shown in the Appendix.

$$W_{WF} = W_{NF} \text{ antilog}_{10}\left(\frac{SPL_{PWF} - SPL_{PNF}}{10}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

where

W_{WF} = acoustic power across the open-duct face with flow (watts).
W_{NF} = acoustic power across the face at no-flow (watts).
SPL_{PWF} = sound-pressure level around the periphery with flow.
SPL_{PNF} = sound-pressure level around the periphery at no-flow.

Since no measurements were made in a moving airstream, the windscreen was not used.

DISCUSSION OF RESULTS

Test results for one of the 4 operating conditions studied are shown in Figs. 6 and 7. At this condition the fan speed was maintained at 1110 rpm with an air delivery of 3600 cfm against a static pressure of 2.4 in. Fig. 6 compares the results of the 4 test methods in terms of watts of acoustic power (left-hand ordinate scale) for the discharge of the fan, and Fig. 7 compares the results of the methods on the inlet side.

To compare the results on an acoustic-watts basis only could be misleading to the layman, since one noise that has double the acoustic output of another would give only a 3 db higher reading. It should also be remembered that measurements of the same source with 2 different instruments could give differences of 2 db due to inherent instrumental errors. For convenience of comparison, therefore, the test results have also been expressed on a power-level basis in decibels for each octave band as indicated on the right-hand ordinate scales in Figs. 6 and 7. By definition

$$PWL = 10 \log_{10} (W/10^{-13}) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

where

PWL = power level (decibels).
10^{-13} = a reference base (watts).

Free-Field and Reverberant Room Methods: The free-field and reverberant-room methods measure the sound power radiated into the space in which the test duct is placed. The radiated acoustic power is dependent upon the acoustic output of the fan and the size and location of the duct opening. In most of the frequency range studied, the 2 methods show good agreement. On the power-level basis, the average difference obtained by these 2 methods for all 4 operating conditions studied was less than 1.5 db for the discharge side in the first 5 octave bands, and about 2 db in the last 3 octave bands. For the study of inlet noise, the average differences obtained by the free-field and reverberant-room methods were 3.5 db in the first band, less than 3 db in the second through sixth bands, and 4 db and 5.5 db in the seventh and eighth bands, respectively. Differences in the first band were probably due to 2 factors. First, it was difficult to measure accurately the background noise in the free-field test. Most of the background noise was concentrated in the first band and fluctuated between −12 and 0 db of the level being measured. The second factor was the uncertain effect of the reactive impedance of the reverberant room at low frequencies.[6] The differences in the last 2 bands (the free-field results being above those of the reverberant room) were probably due in part to the misorientation of the microphone during the free-field studies, or the error arising from the temperature correction. Correction for background noise was necessary only in the first 2 octave bands during the free-field studies. In the reverberant-room tests no background-noise correction was necessary.

The peaks at the 600–1200 cps band in Fig. 7 and at the 1200–2400 cps band in Fig. 6 result from the fundamental frequency and the harmonics of the fan blade impulses. This equals the rpm × number of blades/60 = 1110 × 36/60 = 666 cps, and multiples of this value.

While the free-field test is considered the fundamental method for measuring the acoustic power of equipment, it does require a large open space, (or anechoic chamber) free from reflecting surfaces and with a minimum of background noise. Num-

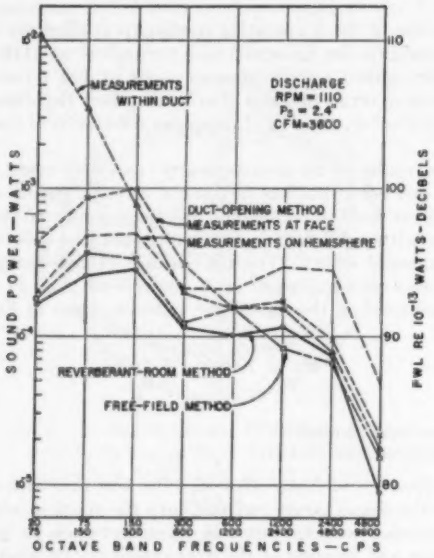

FIG. 6—SOUND POWER VS FREQUENCY-DISCHARGE; POWER LEVEL VS FREQUENCY

erous readings are necessary over the surface of measurement when the noise source has directional qualities. In the reverberant room, the noise source must be acoustically *small* compared to the room. The surfaces must be sufficiently hard to give a reverberant field within the dimensional limits of the space, and the geometry of the surfaces must be such as to diffuse the sound field and to minimize standing-wave patterns. A completely diffused field is difficult to achieve, hence numerous readings or a number of space-average measurements (obtained by swinging the microphone in a circle) may be necessary in order to obtain an accurate reading in the space. The standing-wave effect is greatest in the lower frequencies where the wavelength approaches the room dimensions. To show the fluctuations of sound levels in the space, the condenser microphone was swung on a 3-ft radius circle at 2 different locations in the room, and recordings with the one-third octave-band analyzer were taken at one-third octave-band center fre-

quencies of 80, 160, 320, 640, and 1250 cps. The fluctuations in readings over the circle were 10, 5, 3, 2 and 0 db, respectively. These fluctuations as measured by octave-band filters would be of the order of 3, 2, and 1 db in the first 3 bands.

The work to date indicates that the results obtained by the reverberant room method and by the free-field method might be expected to agree within ±3 db.

Measurements within Test Duct: While measurements within the duct (with horn and termination in place) were made to determine the acoustic power delivered by

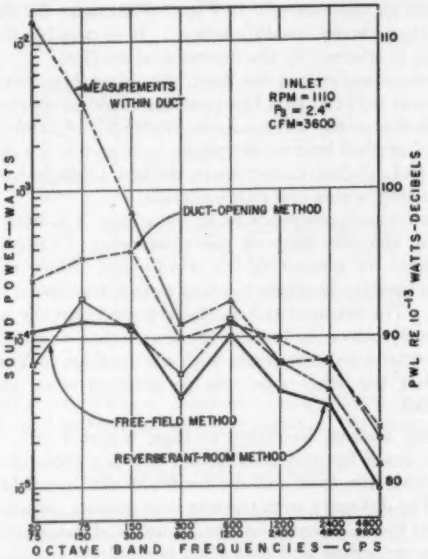

FIG. 7—SOUND POWER VS FREQUENCY-INLET; POWER LEVEL VS FREQUENCY

the fan to the duct system, there are 2 schools of thought as to the exact significance of these readings. One school maintains that although at lower frequencies the energy level in the duct is greater than that radiated from the open end, because of the reflected sound waves resulting from the imperfections of the termination, the actual acoustic power generated by the fan (at either the inlet or discharge) is that which leaves the duct opening and is measured in the reverberant room or free field. The acoustic power must leave in this manner since the duct system is not able to absorb any reflected sound power.

The other school maintains, however, that the measurements within the duct actually indicate the acoustic power delivered by the fan to the duct system, and that the power level so determined would not be seriously affected by imperfections in the horn and termination at low frequencies. It is maintained that that portion of the acoustic power reflected from the open end is absorbed in both the

inlet and discharge ducts of the fan. With the horn and termination in place, the reflected sound power is minimized.

In any case, it is evident that at lower frequencies the energy level within the duct is greater than that radiated from the open end.[1] Whether it appears simply as stored energy or actual acoustic energy being delivered by the fan, it is nevertheless present and must be considered in the design of duct systems to ascertain the attenuation needed between the fan and outlet grille in an air-conditioned space.

Referring to Fig. 6 (discharge side) the differences between the in-duct method and the reverberant-room and free-field methods are in agreement with the theoretical end loss from an open duct.[7] In Fig. 7 (inlet side) the differences are from 3 to 6 db higher than theory would predict.[7] It is possible that the impedance of the duct opening is affected by the direction of air flow.

For measurements made within the duct, the microphone was located near the centerline since it was believed that this position would cause the least disturbance to the airstream and give the most reliable readings. A study of the transverse wave patterns, as described later in the paper, indicated a +2 db correction in the 300-600-cps band and a +3 db correction in the 600-1200-cps band in order to give average values existing across the entire section.

Whenever measurements are made in an airstream, it is necessary to know the self-generated noise characteristics of the windstream. Unfortunately no direct means were available for measuring the wind-noise. Since the windscreen was identical to the one used by Beranek, his data were extrapolated to the air velocities used in this study. The extrapolated results indicated that the self-generated noise levels were sufficiently below the level of the source that no corrections were necessary. Further comparisons were made with the findings of Van Niekerk,[8] which again indicated that the wind noise was no problem when measurements were taken inside the duct.

The Duct-Opening Method: Referring to Figs. 6 and 7, the duct-opening test gave results in the lower frequencies that fell between those obtained from measurements made within the duct and the results of the free-field and reverberant-room methods. The difficulty with the test is in making measurements across the duct opening at no flow. Because of the pronounced longitudinal standing-wave patterns along the centerline and extending beyond the opening, the choice of a location for the microphone was not evident. Moving the microphone along the duct centerline from a position slightly within the opening to a position slightly outside the opening gave average readings about equal to those at the face of the opening at 1250 cps and less. At 2500 cps and above, maximum readings occurred at the face. Readings were ultimately made at the face of the opening. The results would indicate however that acoustically the duct extended beyond its opening for about one-third to one-half of its diameter in the lower frequencies. Subsequent readings made on a hemispherical surface around the duct opening (with the assumption that this was the approximate acoustic end of the duct at lower frequencies) gave results in Fig. 6 that were more in line with the results of the free-field and reverberant-room methods at lower frequencies but somewhat higher at higher frequencies. It would appear that in this test method the readings at the duct opening at no flow should be made over the hemispherical surface (the radius of which equals the radius of the duct) in the first 4 octave bands and at the face of the duct in the last 4 bands.

Comparison Tests: For one test the horn (without the termination) was placed on the end of the duct. Measurements in the reverberant field indicated that the

acoustic power radiated to the space was greater by 4 to 5 db (on a power-level basis) in the first 3 octave bands than for the open-end duct. This is to be expected since the end reflection loss from the horn (26 x 26 in.) is less than the duct opening.

For another comparison the 10-ft test duct was removed and the acoustic power radiated from the short section of duct (the opening being flush with the fan-room wall) was determined. The sound power radiated from the short section was greater than that radiated from the test duct in the first, second, and last 3 octave bands. In the first and second bands differences of 5 and 3 db respectively (on a power-level basis) were noted. The greater differences result from the wall acting as an

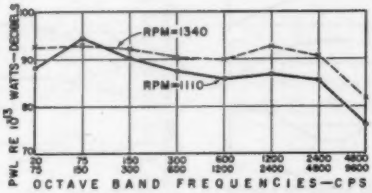

FIG. 8—POWER LEVEL IN FAN ROOM VS FREQUENCY

infinite baffle,[7] and more modes of vibration being excited in the room with the noise source at the wall rather than near the center of the room.

NOISE IN THE FAN ROOM

The sound power radiated from the fan casing, electric motor, and belt drive was measured in the fan room by the reverberant-room method using the standard source. The results on a power-level basis are shown in Fig. 8 for the 2 fan speeds. With the relatively small size of the fan room and the limited positions for measurement, the results are open to question, since insufficient modes would have been excited in the lower frequencies.

STANDING WAVES IN DUCT

Longitudinal Standing Waves: To determine the effectiveness of the exponential horn and *anechoic termination*, the longitudinal wave patterns along the duct centerline were determined with and without the termination in place. This was done by moving the microphone along a track through the length of duct as shown in Figs. 9 and 10. The windscreen was used over the microphone. The results of the wave patterns for the open-end duct are shown in Fig. 9. The wave form showed the greatest variations at the one-third octave band center frequencies of 400 cps and less, and negligible variations at 1600 to 5000 cps. At 640 to 1250 cps a transition was noted between the two extreme conditions. This would substantiate the fact that the duct becomes a less efficient radiator of acoustic power at the lower frequencies.

The longitudinal wave patterns along the duct centerline with the horn and termination in place are shown in Fig. 10. At 400 cps and above the variation along the centerline was negligible. Although the horn has a cutoff frequency of about 41 cps, it becomes a less efficient acoustic radiator at frequencies about an octave above this value.[9] This is shown by the wave patterns at 100, 64 and 50 cps, an indication that a portion of the pressure wave does not reach the termination but is reflected back toward the source. The termination itself becomes less effective at

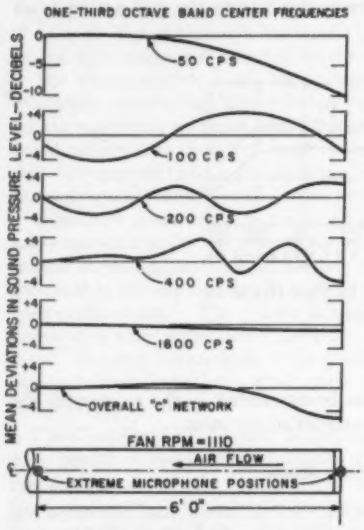

FIG. 9—LONGITUDINAL STANDING WAVES ALONG CENTERLINE OF OPEN-END DUCT

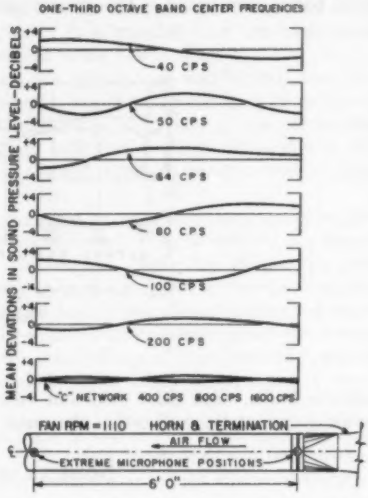

FIG. 10—LONGITUDINAL STANDING WAVES ALONG CENTERLINE OF DUCT WITH HORN AND ANECHOIC TERMINATION IN PLACE

lower frequencies in absorbing acoustic energy. The maximum variations in the wave form with the termination in place were only ±2 db, or one-half as great as the variations occurring in the open-end duct. The wave patterns in Figs. 9 and 10 represent the mean readings of the fluctuations that were recorded. In other words, at any position along the duct centerline, the point-reading would fluctuate with time.

Transverse Standing Waves: At some frequencies the propagation of sound through a duct results in standing-wave patterns normal to the axis as well as in a longitudinal direction. The transverse waves were investigated by positioning the condenser microphone along a radial line outward from the duct centerline. A non-flow condition was used thus eliminating the windscreen. Measurements were made 18 in. inside the open-end duct. The variations relative to the centerline readings are shown in Fig. 11. The principal resonances occurred between

640 and 1000 cps. Measurements made at the centerline alone would therefore give a negative error of about 3 and 4 db in the 300–600 cps and 600–1200 cps octave bands, respectively. Measurements within the duct made at the centerline and 2 in. away resulted in a +2 and +3 db correction (or for the latter a doubling of the acoustic power in watts) for these 2 octave bands as previously indicated.

THE REVERBERANT-ROOM CALIBRATION

The room constants (square feet) are shown for each octave band in Fig. 12. The maximum difference between the values found from the reverberation time and those determined from the standard source is 30 percent. This would be

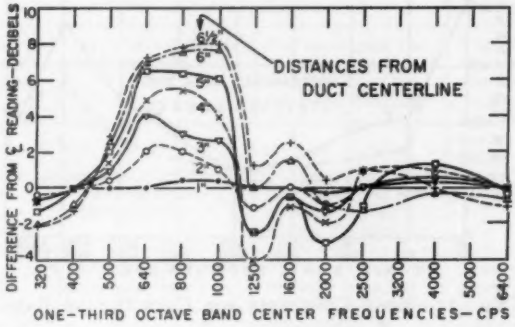

FIG. 11—TRANSVERSE STANDING-WAVE PATTERN IN DUCT

comparable to an error of only 1.5 db which appears to be within the standard error of the instrumentation. The relationship of the room constant to the room area and the average absorption coefficient of the surfaces is given by Equation 7[4], thus

$$R = \alpha S/1 - \alpha. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

where

S = surface area, square feet.
α = average absorption coefficient of the surfaces.

A room constant of 1000 sq ft in the first octave band (shown in Fig. 12) would correspond to an absorption coefficient of about 0.25. From the known properties of the surfaces this value is unusually high. This could result from the difficulty experienced in obtaining readable decay curves in the lower frequencies. The high value of the absorption coefficient determined from measurements made with the standard source could be attributed to gaps in the room response at low frequencies which are due to the small number of room harmonic modes.[9] As a result, some of the sound would not be transmitted to the microphone position in the room.

The increasing values of the room constants (and hence higher absorption coefficients) above the 600 to 1200 cps band result from the increased sound absorption caused by the moisture in the room air.

Although the room constants can be in error in the lower frequencies, it should again be pointed out that this factor is eliminated in the use of Equation 2 for both the unknown and standard noise sources. The room constants have been pre-

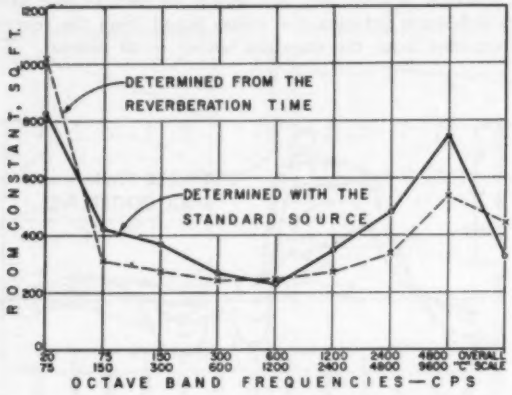
ROOM CONSTANT, SQ FT.
DETERMINED FROM THE REVERBERATION TIME
DETERMINED WITH THE STANDARD SOURCE
OCTAVE BAND FREQUENCIES—CPS
OVERALL "C" SCALE

FIG. 12—ROOM CONSTANTS FOR EACH OCTAVE BAND

sented, however, for the purpose of showing what one might expect to find through measurements for a room of this particular type and size.

APPLICATION OF THE TEST METHODS TO OTHER EQUIPMENT

The test methods studied could apply to other forms of air-moving equipment besides a centrifugal fan. Table 1 outlines the test methods most applicable for determining the acoustic output of various air-moving equipment.

ACKNOWLEDGMENT

The author wishes to thank the Brush Electronics Co. and the General Radio Co. for the loan of sound-measuring equipment.

GENERAL DISCUSSION

No correlation between the sound power of the inlet and discharge ducts has been made. Whether more of the sound power is transmitted down the inlet duct than down the discharge (or vice versa) is probably a function of the fan wheel and housing configuration.

TABLE 1—METHODS ADAPTED TO DETERMINE THE ACOUSTIC-POWER OUTPUT OF AIR-MOVING EQUIPMENT

TYPE OF EQUIPMENT	TEST METHOD RECOMMENDED	ACOUSTIC POWER MEASURED	REMARKS
Unit Heater	Reverberant Room or Free-Field	The total acoustic power of the source is measured	The unit is self contained with no duct system
Axial Fan Without Duct	Reverberant Room or Free-Field	The total acoustic power of the source is measured	The unit is self contained with no duct system
Propeller Fan Without Duct	Reverberant Room or Free-Field	The total acoustic power of the source is measured	The unit is self contained with no duct system
Axial Fans and Centrifugal Fans with Duct System	Reverberant Room or Free-Field	For measuring the acoustic power radiated from the inlet and discharge ducts	The reverberant-room method could also be employed in determining the acoustic power radiated into the fan-room.
Axial Fans and Centrifugal Fans with Duct System	Measurements within duct	For measuring the acoustic energy within the duct	
Window Air Conditioner or Industrial roof exhauster	Reverberant Room or Free-Field	The acoustic power radiated from the outdoor and indoor sides would be measured separately	The unit should be placed in a wall as normally mounted. Free-Field measurements would be made over a hemispherical surface.
Self contained Unit air conditioner without ducts	Reverberant Room or Free-Field	The total acoustic power of the source is measured	Choice of method would depend upon size of unit as compared to the size and shape of test space

Note: The term free-field, applies to either an unobstructed outdoor space or an anechoic chamber.

Acoustic output has been expressed in this paper as watts, since this is an absolute unit. Conversions can be made to a power-level basis when necessary for ease of comparison. When comparing data on a power-level basis the reference base must always be the same. Although the use of logarithms and anti-logarithms in the foregoing equations may appear bulky and awkward to solve, partial graphical solutions were constantly used which greatly simplified and speeded calculations.

There is still a great need for information that correlates the acoustic-power output of equipment with its operating characteristics and dimensional con-

figuration. When more of such data are available, the engineer will be able to predict and more accurately control the noise levels in air-conditioned spaces.

CONCLUSIONS

1. The reverberant-room test gives results that correlate well with the free-field test.
2. At lower frequencies, the acoustic energy in the duct is greater than that which is radiated from the open end.
3. Because of the transverse standing wave effect in the duct, measurements within the duct made at the centerline should be corrected if necessary in the bands where the resonant frequencies occur.
4. At least one, and in some cases two, of the test methods seem well adapted for determining the acoustic-power output of air-moving equipment as noted in Table 1.
5. While the duct-opening test does not require an elaborate setup, it needs further study before it can be applied as an accurate test method.

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APPENDIX

SOLUTION TO EQUATION 3: $W = 10^{-12} S \left(\text{antilog}_{10} \frac{SPL}{10}\right)$

$$W = (IS)\ 10^{-7}$$

where

W = watts of sound power for a plane or spherical wave

$I = \frac{p^2}{\rho c}$ = sound intensity, $\frac{\text{ergs}}{cm^2\ \text{sec}}$

c = speed of sound, centimeters per second

$\rho = \frac{\text{air density}}{g} = \frac{\text{dynes(sec)}^2}{\text{cm}^4}$

P = sound pressure, dynes per square centimeter

$$SPL = 10 \log \frac{P^2}{(0.0002)^2}$$

$$\text{antilog}_{10} \frac{SPL}{10} = \frac{P^2}{(0.0002)^2}$$

$$P^2 = 4 \times 10^{-8} \text{ antilog}_{10} \frac{SPL}{10}$$

$$\rho c = 42.86 \left(\frac{273}{K}\right)^{\frac{1}{2}} \left(\frac{B}{30}\right) \text{ rayls, (See Ref. 2).}$$

For average conditions $P_B = 29.5$, $T = 296.8$ K, for $t = 75$ F

$$\rho c = 42.86 \left(\frac{273}{296.8}\right)^{\frac{1}{2}} \frac{29.5}{30} = 42.86\ (0.92)^{\frac{1}{2}}\ (0.983)$$

$$\rho c = 42.86\ (0.959)\ (0.983)$$

$$\therefore \rho c = 42.86\ (0.942) = 40.3$$

$$\therefore W = \frac{P^2 \times S}{\rho c} \times 10^{-7} = \frac{4 \times 10^{-8}}{40.3} \times 10^{-7} \text{ antilog}_{10} \left(\frac{SPL}{10}\right)$$

$$\times 927 \text{ cm}^2 \text{ per ft}^2 \left(\frac{S, \text{ ft}^2}{1}\right)$$

$$\therefore W = 0.92 \times 10^{-13}\, S \left(\text{antilog}_{10} \frac{SPL}{10}\right) \text{ where, } S = \text{ft}^2$$

or $W = 10^{-13}\, S \left(\text{antilog}_{10} \frac{SPL}{10}\right)$ the use of a coefficient of 1 instead of 0.92 results in only a 0.34 db error.

SOLUTION TO EQUATION 5: $W_{WF} = W_{NF} \text{ antilog}_{10} \left(\frac{SPL_{PWF} - SPL_{PNF}}{10}\right)$

$$I_{NF} \sim \frac{P_{NF}^2}{\rho c}$$

where

I_{NF} = sound intensity at face, no flow

P_{NF} = sound pressure at face, no flow

or $$I_{NF} = C_1 \frac{P_{NF}^2}{\rho c}$$

but $$P_{NF}^2 \sim P_{PNF}^2$$

where

P_{PNF} = sound pressure at periphery, no flow

or $$P_{NF}^2 = C_2 P_{PNF}^2$$

$$\therefore I_{NF} = C_3 \frac{P_{PNF}^2}{\rho c}$$

where

$$C_3 = C_1 C_2$$

and

$$W_{NF} = C_3 \frac{P_{PNF}^2}{\rho c} A_F \quad \text{(A-1)}$$

W_{NF} = sound pressure at face, no flow

A_F = area of face

Likewise

$$W_{WF} = C_3 \frac{P_{PWF}^2}{\rho c} A_F \quad \text{(A-2)}$$

W_{WF} = sound power at face, with flow

P_{PWF} = sound pressure at periphery, with flow

Dividing Equation A-2 by A-1,

$$\frac{W_{WF}}{W_{NF}} = \frac{P_{PWF}^2}{P_{PNF}^2}$$

$$10 \log \frac{W_{WF}}{W_{NF}} = 20 \log P_{PWF} - 20 \log P_{PNF}$$

or

$$10 \log \frac{W_{WF}}{W_{NF}} = SPL_{PWF} - SPL_{PNF}$$

and

$$W_{WF} = (W_{NF}) \text{ antilog}_{10} \left(\frac{SPL_{PWF} - SPL_{PNF}}{10} \right)$$

DISCUSSION

C. M. ASHLEY, Syracuse, N. Y., (WRITTEN): This paper represents an important contribution to the technique of measuring sound-power output of air-conditioning equipment. The comparison of the 4 different methods indicates that each method has certain areas of special advantage and utility; the reverberant-room method, semi-reverberant-room method, or the free-field method, would appear to be well suited to measurement of radiation from room air conditioners, self-contained air conditioners, outlets, compressors, and similar equipment where we are principally concerned with the noise radiated from equipment to a room.

For self-contained equipment where the principal source of noise is radiation from a duct opening, the duct-opening method has much to commend it from the standpoint of both simplicity and freedom from interference from ambient noise.

For the measurement of noise transmitted through ducts of a central station system from a fan, it appears that there is no choice but to use the duct method of measurement, since at low frequencies the straight duct termination has a very high reflective loss. While some of this reflected energy will be re-reflected from the fan and other end of the duct, even the small amount of dissipation which exists in the system will absorb most of this reflection in the very low-frequency bands and will thus make it impossible to predict the true sound-power output of the fan from measurements made outside the duct.

Unfortunately in the middle-frequency range, there is a transverse standing wave pattern in the duct, but the effect of this can be minimized by testing at the nodal point of these waves. The anechoic termination used in these tests had some imperfections in the lowest 2 frequency bands. But, it is believed that we now know how to eliminate most of the imperfection.

From the results of the study that was made, where the fan is to be used with both inlet and discharge ducts, it would be desirable to provide the test setup with anechoic terminations for both of these ducts. However, where the fan is to be used with an inlet plenum, then it would probably be better to use a duct and anechoic termination only on the discharge with a test room outside of the duct on the inlet.

The reliability of the reverberant-room method in the lowest octave band seems to be open to considerable question. Measured value of the room constant appears to be unreasonable. It is believed that this may be in part due to the gaps between modes in the standing wave pattern of the room. It is most important to emphasize the use of a reflective room of adequate size and optimum proportions or other techniques for averaging the room performance.

This paper should be of great value in providing background information to the ASHAE-*ASRE* Standards Committee which is now being organized to develop air-conditioning noise test standards.

R. L. HULL, Davenport, Iowa: I can give a person some information as to the origin of the nomograph which appeared in the June 1957 issue of *Heating, Piping & Air Conditioning*. The nomograph is the result of tests made by Armour Research Foundation of two commercial fans in their Riverbank Laboratory Reverberation Room. One fan was a propellor type and the other a centrifugal type roof ventilator. Each fan was tested to give sound-power output on the intake side and on the exhaust side. Absorption constants were calculated by measuring the decay time for each of 8 octave bands for the 2 mountings of each fan. The door to the reverberation room was blocked open far enough to allow air flow to be near the middle of the rising portion of the fan performance curves. Sound-pressure levels were read in each of the 8 octave bands. Using these readings and the absorption constants, sound power in microwatts was calculated.

Since the intake and the exhaust powers did not differ very much, the nomograph is based on an average of the power intensity on the surface of a one-foot radius sphere whose center is at an imaginary point source of sound. Then, assuming constant impedance, any measured sound intensity represents a specific sound-pressure level at one foot. As a convenient starting point in solving practical problems, the nomograph uses sound-pressure level at one foot rather than sound power in microwatts.

The two fans tested at Armour have been used as secondary standards to calculate the absorption constants of our reverberation room at E. Moline.

We have tested and are testing a large number of fans of various sizes, types, and tip speeds. The nomograph shown in the article gives a good approximation of the sound-pressure level at one foot of these fans. However, we believe that we may revise the present nomograph or make several nomographs for the several types of our fans to allow calculation of the sound-power curves which will be of even greater accuracy.

Although the tests at Armour did show a break in the sound-power curve in the 37.5–75.0 band, they did not show this band in the nomograph. Thus, omitting the lowest band, a good approximation of the sound-power curve is given by a straight line. Our further tests indicate a break in the sound-power curve which is related to the blade frequency.

AUTHOR'S CLOSURE: Mr. Ashley's discussion points out some of the difficulties of the in-duct test method. At the mid-frequency range where transverse standing-wave patterns were noted, the arithmetic mean values of the sound-pressure levels, measured radially outward from the duct centerline, were used in correcting the centerline reading to represent an average level in the duct. While this is not an absolute approach to the problem, it did give results that were in line with the other test methods. As Mr. Ashley

indicated, a position for the microphone location will have to be chosen so that the error is minimized compared to readings taken at the duct centerline only. As indicated, the anechoic termination used in the study was imperfect in the lower frequencies. This certainly points to the need for developing an effective and near-perfect termination if the in-duct test method is to be incorporated in a standard test code. Since a termination was used on only one end of the duct, reflections from the opening of the other end could have caused an error in the in-duct measurements. Mr. Ashley is correct in pointing out that an anechoic termination should be used on both the inlet and discharge ends of the duct when the fan inlet and discharge are normally coupled to a duct system.

In the article outlined by Mr. Hull, I recall that a table is given whereby the sound-pressure level of the fans tested dropped off at the rate of 5 db per octave in the 150–300 cps band and above. This drop in level as the frequency increases is also noted in in-duct measurements of centrifugal fans when determining the acoustic power delivered by the fan to the duct system. Since the commercial fans used in the article were tested as roof ventilators, their inlet and discharge ducts were probably short sections, and measurements of their acoustic output were made beyond the duct openings in the reverberant field. It is interesting to note this similarity of results. In regard to the nomograph mentioned, at a fixed tip speed of about 6,500 fpm, the sound-pressure level of the fan at 1 ft. in the 75–150 cps band (which is about 1.5 db less than the over-all level) is as follows: $SPL = 10 \log hp + 100$. It is again interesting to note that this is identical to the relationship developed by Beranek and Allen for the over-all power level of a fan.

It is stimulating to see increasing interest in air-conditioning noise and noise measurement in general. The future for noise control looks bright. Not only will the engineer have better methods and procedures for measuring the noise output of equipment, but data should be available for predicting the acoustic output over a wide range of operating characteristics.

1610

No. 1610

ACOUSTIC POWER LEVEL DETERMINATION FOR MACHINERY IN HARD ROOMS†

By W. J. BURCH*, ANNAPOLIS, MD.

IT HAS long been the practice to write machinery noise specifications in terms of sound-pressure-level (SPL) measured at specified distances. This has objectionable limitations since the SPL depends not only on distance, but also upon the acoustic characteristics of the test location, *i.e.*, room size, acoustic treatment, and others. Since it is extremely difficult to duplicate exact room acoustic characteristics, significant discrepancies have been noted from time to time in the results obtained by different test facilities. From experience of this sort, it has become increasingly apparent that there is need for a noise level criterion which does not depend on the acoustic environment of the test location. One such criterion is acoustic power or acoustic power level (PWL). It is also desirable to use acoustic power or PWL as the parameter for specification purposes because it is a quantity from which sound-pressure-level can be predicted at various distances and locations with respect to the equipment regardless of the place where the PWL was determined.

This project was undertaken to determine the practicability of calculating the PWL of a piece of machinery, located in hard or semi-hard rooms, using the SPL measured in the diffuse field and certain estimated room constants. In addition, it was desired to determine if the type of machinery mounting would cause significant changes in PWL. Also, it was desired to know, as a matter of practical experience, how accurately room constants could be estimated.

MACHINERY AND SOUND SOURCES INVESTIGATED

Three different sound sources, identified as follows, were used in the investigation:

1. A small 1/6-hp, electric motor driven, household type refrigerator compressor, with suction and discharge open to atmosphere.

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† The opinions expressed in this paper are those of the writer and do not necessarily reflect the views of the Navy Department or of the naval service at large.

* Acoustics Branch, Noise, Shock and Vibration Laboratory, U. S. Naval Engineering Experiment Station.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Murray Bay, Que., Canada, June 1957.

2. A 3-hp, 2-stage, electric motor driven air compressor with suction and discharge open to atmosphere.
3. A white noise generator and amplifiers, frequency range 50 to 6500 cps.

NOISE INSTRUMENTATION

Principal instruments used in the investigation included the following: (*a*) Type 1551–A sound level meter, (*b*) Type 3A frequency analyzer, (*c*) Type 633–A microphone, (*d*) EES impedance matching transformer, (*e*) Type 2304 high speed sound level recorder, (*f*) Type 2109 audio frequency spectrometer, and (*g*) Type 21B microphone.

All sound-level measurements were made with a Type 633–A microphone coupled to either the Type 3A analyzer, or Type 1551–A sound level meter. An EES designed transformer was used to match the impedance of the 633–A microphone with the sound-level meter.

Measurements of reverberation time were made with the Type 21B microphone coupled to the Type 2304 high-speed sound-level recorder through the amplifier section of the Type 2109 audio-frequency spectrometer. The diffuse noise fields for these measurements were established by use of both a discrete frequency warble tone oscillator and a white noise generator-amplifier unit.

The microphones used were calibrated, for selected frequencies in the range covered, by a substitution method using 3 reciprocity-calibrated Type 640AA condenser microphones as standards. The measurement systems were electrically calibrated at the start of each series of measurements and periodically at any other time there was reason to suspect their accuracy.

METHOD OF TEST

The first phase of the experiment was to calculate the acoustical power of the noise source in an anechoic chamber by spherical integration. In this case, the sphere was divided into 24 equal areas. The geometrical center of each area was located and the SPL was determined at this point.

The SPL was then calculated as the arithmetic average of the sound intensities. Acoustic power then is

$$W_a = 10^{-13}\, S\, [\text{antilog}_{10}\, (\text{SPL}_{ave}/10)] \text{ watts} \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where

S = area of the incremental area in square feet.

SPL_{ave} = average sound-pressure-level.

Total acoustic power is then

$$W = W_a \times n \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

when

n = number of incremental areas.

The total sound power W in watts as expressed by the power level is

$$\text{PWL} = 10 \log_{10} W + 130 \text{ db} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

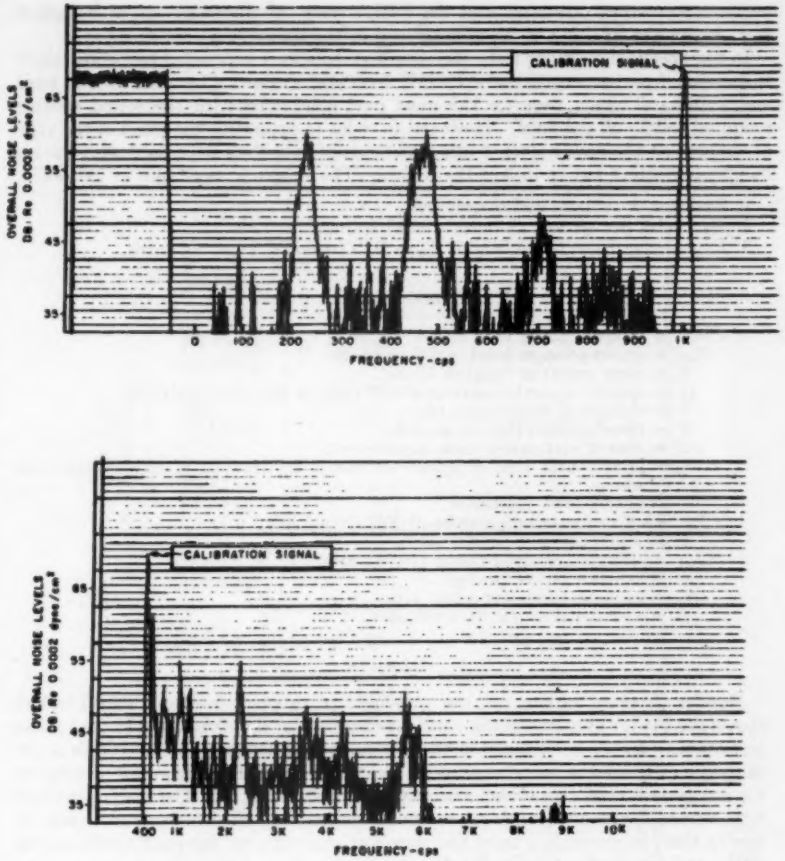

FIG. 1—TYPICAL FREQUENCY SPECTRA OF HOUSEHOLD REFRIGERATION COMPRESSOR (AIR-BORNE SPECTRA)

The second stage of the experiment involved determination of the powers in various rooms. However, before these could be determined, it was necessary that a room constant or acoustical index be established for each of the individual rooms. This was accomplished by 2 methods. First, the reverberation times were accurately measured with the high speed level recorder and from these the room constants were calculated by Equation A-2, Appendix A. Second, the areas of each type of material in the rooms were either measured or estimated and from published absorption coefficient data the average absorptivity and room constants were determined by Equations A-3 and A-4, Appendix A. The results of these 2

methods were then compared and the PWL's were calculated by use of Equation A–1, where the SPL is the value measured in the diffuse noise field.

In the anechoic chamber tests, the sound sources were supported by rubber band suspension from an acoustically treated overhead crane. The tests in the hard and semi-hard rooms were for the most part conducted with the sound sources mounted solidly on the floor. However, in order to determine the effect of the type of support on the power level, 3 other methods of mounting were used with the air compressor sound source in 3 of the rooms. The supports used were as follows:

1. Resilient or rubber band suspension.
2. Six cycles per second, cylindrical steel springs.
3. Engineering Experiment Station Type B7S (nominal 6 cps) mounts.

NOMENCLATURE

PWL = acoustic power level re: 10^{-13} watt.
SPL_d = sound-pressure-level in diffused field.
R = room constant (English Units).
ρc = specific acoustic impedance (407 rayls or 40.7 mks rayls/ft²).
V = volume of room (cubic feet).
T = reverberation time in seconds.
S = area of surfaces of room, square feet.
S_n = incremental area of sphere at radius of distance of microphone from source.
a = absorptivity coefficient.
P_{ref} = reference sound pressure (0.0002 dyne/cm²)
P = sound pressure (dynes/cm²)
W_{ref} = reference acoustic power (10^{-13} watt).
W = acoustic power (watts).
W_a = acoustic power for a unit area.
A = average absorptivity coefficient.

RESULTS

Results obtained in the test are presented in tabular form as Table 1, which shows the PWL's obtained in the anechoic chamber *vs.* those obtained in the various test rooms. SPL's determined in the test rooms are also presented. Table 2 presents the estimated *vs.* calculated room constants. Sound spectrum for a refrigerator compressor used as a sound source is presented as Fig. 1. The results obtained with various types of mountings are not presented numerically; however it was observed that, in proceeding from the very soft mounting to the rigid condition, the PWL was constant within ± 0.5 db.

DISCUSSION

The data presented in Table 1 show that the PWL's determined in the test rooms were generally lower than those found by spherical integration in the anechoic chamber. The reasons for this are not exactly known; however, nonsymmetry of the sound sources, possible imperfect anechoic chamber characteristics, inherent error due to instrumentation, and specific acoustic phenomena associated with the test rooms could all be contributing factors to the discrepancies noted. On the other hand, the PWL's obtained in the majority of test rooms agreed to within ±2.0 db of the *average* value.

The data presented in Table 2 show that good agreement was found between the estimated and calculated room constants. The maximum discrepancy in these

TABLE 1—TABULATED RESULTS OF TEST

PWL = DB RE: 10^{-13} WATT SPL = DB RE: 0.0002 DYNE/CM^2

NOISE SOURCE	AVG PWL ANECHOIC CHAMBER	HALLWAY WAVE MECH LAB	CALIBRATION ROOM WAVE MECH LAB	VESTIBULE WAVE MECH LAB	SHOCK & VIBRATION OFFICE WAVE MECH LAB	LAUNDRY ROOM BANCROFT HALL USNA	STOREROOM 104 BANCROFT HALL USNA	SQUASH COURT BANCROFT HALL USNA	CLASSROOM BUILDING 120 EES	HONOR ROOM BANCROFT HALL USNA	ROOM 103 BANCROFT HALL USNA	CREW'S MESS USS TURBOT	AFT TORPEDO ROOM USS TURBOT	
1/6 hp Refrigerator Compressor	80.0	79	77	77	81	79	77	78	—	82	79	—	—	PWL[a]
		76	81	77	81	80	78	78	—	81	79	—	—	PWL[b]
		69	67	78	63	68	61	61	—	66	65	—	—	SPL_d[c]
3 hp Air Compressor	102.0	98	95	—	—	—	—	—	—	—	—	—	—	PWL[a]
		95	99	—	—	—	—	—	—	—	—	—	—	PWL[b]
		88	84	—	—	—	—	—	—	—	—	—	—	SPL_d[c]
White Noise Generator	99.0	94	90	96	96	97	96	95	97	94	95	95	98	PWL[a]
		91	94	96	96	98	97	95	96	94	95	100	97	PWL[b]
		85	79	97	78	86	80	78	76	77	81	77	80	SPL_d[c]

[a] PWL Estimated: PWL calculated from the diffuse field SPL and the estimated room constant.
[b] PWL Calculated: PWL calculated from the diffuse field SPL and the room constant calculated from the measured reverberation time.
[c] SPL_d: The diffuse field SPL.

values was approximately 20 percent, with the exception of those few rooms which have questionable acoustic properties. The large discrepancy in the hallway values was probably caused by the 6-ft false ceiling space and oblong dimensions which tend to cause nongeometric acoustics. No explanation has been found for the large variation noted in the calibration room. The crew's mess compartment of the hulk

TABLE 2—TEST ROOM ACOUSTIC ABSORPTION CHARACTERISTICS

TEST ROOM	VOLUME OF ROOM FT³	AREA OF ROOM SURFACES FT²	AVERAGE ABSORPTIVITY COEFFICIENT	ESTIMATED ROOM CONSTANT	MEASURED REVERBERATION TIME, SECONDS	CALCULATED ROOM CONSTANT
WAVE MECH LAB						
Hallway	400	350	0.105	41	1.00	21
Calibration Rm	3,630	2,320	0.020	47	1.50	116
Vestibule	200	226	0.020	4.6	2.20	4.6
Shock and Vibration Off.	4,650	1,702	0.145	291	0.85	295
BANCROFT HALL, USNA						
Laundry Rm	4,600	1,820	0.032	61	3.60	65
Rm. 103, basement	2,756	1,617	0.058	99	1.40	101
Honor Rm, basement	3,610	1,575	0.113	200	1.10	172
Rm 104, basement	5,900	2,881	0.052	158	1.55	196
Squash Court	10,700	3,014	0.072	233	2.30	242
BUILDING 120, EES						
Classroom	6,000	2,264	0.181	500	0.80	406
USS TURBOT						
Crew's Mess	3,910	3,015	0.081	267	0.60	833
Aft Torpedo Rm	4,590	2,245	0.111	286	1.10	218

of a typical submarine contained many partial partitions and a storage space enclosed by wire mesh. Its length was also several times its height or width. In view of these results it appears that rooms such as these, as well as rooms where the principal dimensions are not of the same order, should not be used because of the resulting nongeometric acoustics and poor diffusion of sound. However, if it becomes necessary to use a room of this kind, the reverberation time should be measured and the room constant calculated therefrom. The data also indicate that it is not necessary to use an empty or nearly empty room in order to obtain satisfactory results. In the rooms tested, essentially the same estimated room constants were calculated by at least 2 people working independently.

OTHER OBSERVATIONS

It is realized that the number and size of sound sources used in the investigation were limited and that further studies involving larger and more varied pieces of equipment should be made. The limitations were imposed chiefly by considerations of portability and power requirements. These reasons also led to choice of the amplifier-speaker system as one of the sound sources.

Tests were conducted with the latter source using pure tones (both steady and warbled), but no satisfactory power could be calculated due to the standing wave patterns which were present. The tendency for standing waves to appear was also

noted when the air and refrigerator compressors were used. Under given conditions, the air compressor produced a projected pattern which varied by approximately ±1.0 db, whereas the refrigerator pattern varied as much as ±2.0 db.

The location of the noise source in the room had no noticeable effect upon the diffuse SPL reading.

CONCLUSIONS

The powers determined in the test rooms were in general lower than those found by spherical integration in the anechoic chamber.

The powers determined in the test rooms agreed to within ±2.5 db of the *average* value. The maximum decibel spread was 5 and 9 db (PWL). Corresponding spreads in sound-pressure-levels measured were 17 and 21 db.

The estimated room constants obtained were within approximately 20 percent of the values calculated from the measured reverberation times. This made a difference of less than 1.0 db in the PWL calculation and indicates that in most cases it will not be necessary to establish room constants by making reverberation time measurements.

The type of support used for the noise source did not significantly affect the calculated PWL's. In proceeding from the very soft mounting to the solid or rigid condition, the PWL varied by only 0.5 db.

The white noise sound source did not produce detectable standing wave patterns, while the refrigerator and air compressors produced standing wave patterns which varied ±2 and ±1 db, respectively.

ACKNOWLEDGMENTS

Credit is given to Lt. C. E. Rakes, USN and Lt. (jg) J. E. Kaune, USN, who performed this work during summer recess from training duty at Massachusetts Institute of Technology.

APPENDIX A

EQUATIONS USED FOR CALCULATIONS

Power Level in Reverberant Room:

$$\text{PWL} = \text{SPL}_d - 6.0 + 10 \log R - (10 \log \text{pc} - 15.7) \quad \text{(A-1)}$$

Measured Room Constant:[1]

$$R = 0.05\ V/\text{T}\ [1 + (0.025\ V/TS)] \quad \text{(A-2)}$$

Room Constant:[1]

$$R = \text{A}\ S/(1 - \text{A}) \quad \text{(A-3)}$$

Estimated Average Absorptivity:[1]

$$\text{A} = (a_1S_1 + a_2S_2 + a_3S_3 + \ldots)/(S_1 + S_2 + S_3 + \ldots) \quad \text{(A-4)}$$

Power Level in Free Field:[2]

$$W_s = 10^{-13}\ S_n\ [\text{antilog}_{10}\ (\text{SPL}/10)]\ \text{watts} \quad \text{(A-5)}$$

$$\text{PWL} = 10 \log_{10} W + 130\ \text{db} \quad \text{(A-6)}$$

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[1] *Handbook of Acoustic Noise Control*, by Richard H. Bolt (Physical Acoustics, WADC Technical Report 52-204, Vol 1, December 1952, Equations A-1, A-2, A-3, A-4, pp. 76–79).

[2] *Acoustics*, by Leo L. Beranek (McGraw-Hill Engineering Series, Equations A-5 and A-6, pp. 366–368).

APPENDIX B

RECOMMENDED PROCEDURE FOR THE CALCULATION OF PWL IN HARD AND SEMI-HARD ROOMS

Estimate the room constant by the equation:

$$A = \frac{\sum_1^n a S_n}{\sum_1^n S_n} \qquad \text{(B-1)}$$

a = absorption coefficient of each kind of material.

S_n = area of each surface.

A = average absorptivity.

and the equation $R = A\,S/(1 - A)$

$$S = \text{total area}\left(\sum_1^n S_n\right) \qquad \text{(B-2)}$$

To determine the SPL *in the diffuse field:*

First determine the point at which direct field noise ceases to predominate or where the SPL per doubling of distance no longer decreases by approximately 6.0 db. This point is usually several source diameters from the noise source but varies depending on the amount of absorption material present in the room. Starting from this point an arbitrary 10 ft long straight line should be traversed, taking a reading every foot. No point on this line should be less than 2 ft from a wall, floor or any solid object nor should any point be less than $0.282\sqrt{R}$ (R = room constant) from the piece of machinery being tested. The average of the readings is the diffuse SPL.

To determine the PWL *use the equation:*

$$\text{PWL} = \text{SPL} - 6 + 10 \log R - (10 \log pc - 15.7) \quad . \; . \; . \quad \text{(B-3)}$$

The portion of this equation which is in parentheses is usually less than a decibel and may be neglected in normal engineering work.

APPENDIX C

SAMPLE ESTIMATED ROOM CONSTANT CALCULATION

Sample estimated room constant calculation aft torpedo room:

MATERIAL	ABSORPTION COEFFICIENT a	AREA S_n (SQ FT)	SABINS $a = aS_n$
Steel (transverse blhd)	0.02	200	4.0
Corkboard (side blhd)	0.30	635	190.0
Steel (side blhd)	0.02	272	5.4
Steel (deck)	0.02	518	10.4
Miscellaneous Wood	0.10	200	20.0
Miscellaneous Steel	0.02	300	6.0
Steel Cable	0.03	200	6.0
Screen Cage			7.0
Tank (plaster)	0.04	20	0.8
		ΣS 2345 = S	Σ_a = 249.6 = ΣA

$A = \Sigma a/\Sigma S = 0.106$

$1 - A = 0.894$

$R = S A/(1 - A) = 249.6/0.894 = 278$

Determination of calculated room constant:

T = reverberation time = 1.1 seconds.

$V = 4590$ ft³.

$R = 0.05\ V/T\ [1 + (0.025\ V/TS)]$.

$R = [4590\ (0.05)/1.1]\ \{1 + [0.025\ (4590)/1.1\ (2345)]\}$.

$R = 218$.

Determination of PWL:

SPL = 80.0 db.

SPL = PWL $- 10 \log R + 6.0 + (10 \log \rho c)$.

Estimated value PWL (using estimated R):

PWL = 74 + 24.2 = 98.1.

Calculated value PWL (using calculated R):

PWL = 74 + 23.1 = 97.1.

DISCUSSION

W. F. KERKA, Cleveland, Ohio: The author is to be commended for his clear summary of the methods for estimating and measuring the reverberant characteristics of spaces, and of comparing the power levels of noise sources in these spaces with those levels found under free-field conditions.

While the measurement of reverberation time for the over-all frequency spectrum does not impose many problems as far as accuracy is concerned, the measurement of decay rates at the lower frequencies becomes increasingly difficult. A simple device such as a toy paper-breaker gun will give very clear decay curves down to about 1000 cps, but below this frequency, it is difficult to find mechanical devices giving sufficient low-frequency impact noise that is well above the background level of the space. Hence the low-frequency decay curves recorded on a high-speed sound-level recorder become rather jagged and hard to measure.

I should like to ask Mr. Burch if any decay rates were made at the lower frequencies and if any difficulties were experienced?

C. M. ASHLEY, Syracuse, N. Y.: The study reported by Mr. Burch gives further confirmation of the ability to predict sound power level from room sound-pressure-level. The value of the work is somewhat diminished by the failure to report the measurements in terms of frequency band. The frequency spectrum shown in Fig. 1 is deficient in noise below approximately 200 cycles per second, where most of the difficulties with the correlation of sound-pressure-level and sound power level can normally be expected to lie.

It would be helpful to learn from Mr. Burch whether this deficiency also exists for the other sound sources measured. Since certain types of air-conditioning and refrigerating equipment have important low frequency sources of noise, this frequency deficiency in the sources measured takes on added importance.

AUTHOR'S CLOSURE: With regard to Mr. Ashley's comment it is realized that if the measurements had been made in terms of frequency band it would be reasonable to expect somewhat more spreading in the data. Also, the spectrum presented in the paper is typical of the other 2 sources used, as none of the sources exhibited strong frequency components much below 200 cps.

In reply to the point raised by Mr. Kerka, we did not employ a mechanical or impact type noise source in the measurement of room reverberation time. Rather, we used the

B and K measurement system which consists of an oscillator driven loudspeaker system. Provisions are made for driving at discrete frequencies, but with a warble tone for better diffusion of sound. Also, the oscillator is equipped with a silent switch for instantaneous interruption of the signal. The standard frequencies from 125 through 4,000 cps were used. No difficulty was experienced in making the measurement at the lowest frequency because good dynamic range above background could be obtained.

1611

No. 1611

STUDIES IN SILENCING A DIESEL DRIVEN HEAT PUMP

By H. H. VICKERS* AND R. W. SAGE**, LINDEN, N. J.

WIDESPREAD interest in some form of summer cooling has been amply demonstrated by the expanding sales of room air conditioners and by the rapid development of a market for central air-conditioning units even for small homes. The central units usually consist of conventional cooling units and attached furnaces, or of the all-electric heat pump. However, in the colder areas where the heat pump suffers from lower efficiency as the outside temperature drops, the investment and operating costs of the all-electrical system are normally higher than competitive systems. This is because resistance heating is used to supplement the heat pump output rather than size the heat pump itself to provide all the heat at the lower temperature.

However, the heat pump is not limited to an electric drive as others have pointed out. In this study, a diesel engine drive has been adapted to an otherwise *normal* heat pump. This engine, incorporating speed control and exhaust heat recovery, can handle a larger heating load relative to the cooling requirement than can the constant-speed electric drive.

The study concentrated on the technical difficulties that would have to be overcome before the diesel engine could be seriously considered for home use. It had to be made fully automatic and quiet. The noise problem seemed more difficult since conventional diesel engines are frequently noisy and would be totally unacceptable in the basement of the average home. Therefore, the work discussed herein traces the steps taken in adapting a diesel engine to a heat pump and reducing the noise from an intolerably high level to a murmur, even quieter than a quiet oil burner.

APPARATUS AND TESTS

The heat pump used was a standard commercial 1954 5-ton unit, air-to-air machine. It uses Refrigerant 12 and has a 5 hp compressor motor with a belt-

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Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Murray Bay, Que., Canada, June 1957.

driven compressor operating at 875 rpm. It consists basically of a compressor, condenser coil, expansion valve and evaporator coil with transfer valves to reverse the cycle when needed. Indoor air is forced over one of these coils to either heat or cool the house and outdoor air is forced over the other to either pick up or give out heat. The heating capacity of this unit varies from 70,000 Btu per hr at 50 F to 34,000 Btu per hr at 0 F. If more heat is needed at low temperatures, it is normally supplied automatically by electric resistance heaters.

The diesel engine selected was an air-cooled 4-cycle single-cylinder engine with 3.15-in. bore, developing 5 hp at 1500 and 6 hp at 1800 rpm. It weighed 390 lb and had a fuel consumption of 0.47 lb per hp-hr under the test conditions at full load. The engine was equipped with a 12 volt battery, starting motor, generator and an oil filter.

The automatic controls were battery operated and contained all the relays, switches and timing elements for automatically starting and stopping the engine. A thermostat automatically starts the diesel engine and heat pump or reverses the heat pump cycle to supply either hot or cold air and maintains the desired temperature within 2 deg of any manual presetting.

Although an air-cooled engine was used, later studies indicate that engine heat might be used more effectively if picked up by a water jacket and delivered to a radiator. A water-cooled engine should also be quieter than an air-cooled engine due to the noise absorbing effects of the water and cooling jacket wall and the elimination of sheet metal air cooling ducts.

Before discussing in greater detail the tests that were carried out in silencing the diesel engine, it might be well to review the reasons for choosing the diesel prime mover and the advantages that could exist over an electrical drive. They are as follows:

1. Lower energy cost in most areas.
2. Automatic speed control.
3. Waste heat from the engine can be used instead of supplemental resistance heating to reduce the size of the compressor, indoor and outdoor coils, and both air blowers.
4. Can be adapted to provide hot water.
5. Can be made totally independent of outside power.
6. Can deliver heat at a higher level due to the high level heat of the engine cooling system.

The output power cost for a diesel engine at 28 percent efficiency with 14 cents per gal fuel is 0.36 cents per 1000 Btu *vs.* electric power costs at 2 cents per kwhr of 0.60 cents per 1000 Btu.

The load as well as the heating capacity of the usual constant speed electric heat pump varies with outside temperature. Paradoxically, at low temperatures when the heat demand is greatest, the load on the compressor is smallest, due to low evaporation rates. A diesel engine with speed control, would give greater output at low temperatures resulting in a smaller design size and give modulated output when either heating or cooling.

The heat generated by a diesel engine can be collected, preferably with a water jacket on the engine and exhaust, and delivered to a radiator in the indoor warm air

ducts. The total heat available from this source is 72 percent of that present in the fuel, 28 percent going to mechanical energy. The 28 percent to mechanical energy is used by the heat pump with a coefficient of performance (CP) of 2.2 at 0 F out-

TABLE 1—TYPICAL RESULTS OF PERFORMANCE TESTS

	COOLING CYCLE	HEATING CYCLE
Engine speed, rpm	1400	1200
Rated hp	4.6	4.0
Compressor speed, rpm	548	470
Outdoor air to heat pump, F deg	93	57
Outdoor air from heat pump, F deg	—	47
Exhaust air from heat pump, F deg	103	—
Room air to heat pump, F deg, dry-bulb	72	73
Room air to heat pump, F deg, wet-bulb	62	61
Air from heat pump to room, F deg, dry-bulb	57	110
Air from heat pump to room, F deg, wet-bulb	53	71
Relative humidity, in, percent	57	49
Relative humidity, out, percent	78	12
Hot water from exhaust jacket, lb/hr	100	100
Water ΔT, F deg	68	70

HEAT BALANCE

In: fuel used	Kerosene	Heating oil[a]
lb/hr	1.87	2.01
Btu	37,000	39,200
Out: sensible heat	23,000	50,000[b]
latent heat	14,200	—
heat to water	6,800	7,000
Total	44,000	57,000
Coefficient of Performance	3.2	5.6
Oil Efficiency—%	101[c]	146

[a] No. 2 heating oil. [b] To air. [c] Cooling only.

door temperature for a return of 61 percent of the total energy in the fuel. If 85 percent of the remaining 72 percent can be utilized, the only loss being the cooled exhaust, then the return from the cooling jackets would be 60 percent of the total energy in the fuel. Therefore, the heat recovered from the engine jackets is equal to the output from the heat pump. Thus, for a given cooling capacity, the heating output can be increased to nearly double that of the usual electrically-driven heat pump without resorting to resistance heating.

The engine can also drive a generator to provide power for operation of the air circulating fans or direct pulley drives could be arranged. Either of these methods,

coupled with the battery operated engine controls, makes the unit completely independent of outside power sources.

The noise level of the silenced diesel engine alone proved to be half that of the present heat pump. If advantage is taken of all possibilities the diesel-driven heat pump would be as quiet as a good oil burner, with a noise level competitive with an electric-driven heat pump of the same capacity.

A number of performance tests were made during these studies. Typical results are summarized (Table 1).

These data were taken with both the air inlet and outlet in the same room and some mixing no doubt occurred; however, these data check well with that expected and with the manufacturer's published capacities after allowance is made for the

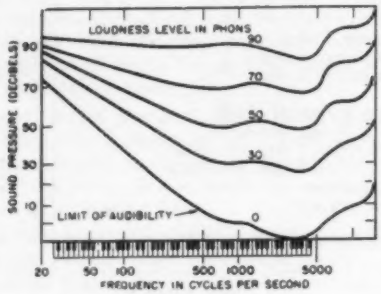

FIG. 1—CONTOURS OF EQUAL LOUDNESS (PHONS)

different compressor speeds. More extensive tests would be required to check the exact value of the hot engine cooling air being delivered to the out-door coil during the heating cycle.

No difficulties were encountered in several hundred hours of operation with either the heat pump or the diesel engine.

ACCEPTABLE NOISE LEVEL AND GENERAL ASPECTS OF NOISE

Before attempting to reduce the noise level of the diesel engine, an acceptable noise level had to be established. A noise level was required for the engine that would not be annoying or, if possible, would not be heard through the floors or walls of a private house in a quiet residential area. This required consideration of the general aspects of noise because sound pressure in decibels is not directly related to the annoyance it can cause.

Loudness is a subjective quantity measured by the human ear. It is a function of the intensity of sound but is complicated by other factors. It is also dependent on continuity and frequency. Intermittent noises common to most internal combustion engines attract attention and require special consideration, but advantage can be taken of the idiosyncrasy of the human ear where loudness varies with frequency.

Loudness level contours have been established[1] from a statistical analysis of hearing tests, in which many people matched the level of one tone to the loudness of another. These *Fletcher Munsen* loudness contours, used to convert decibels into phons or loudness units, are illustrated in Fig. 1. Phons are equal to decibels at 1000 cps but vary at other frequencies depending on intensity. Advantage was taken of this phenomenon by giving more attention to the critical regions of the sound spectrum. For instance, the soundproofed diesel engine has a sound level of 83 db, but has a loudness level of only 58 phons.

Hale Sabine converted equal loudness contours into equal annoyance contours[2] as shown in Fig. 2. By comparing the sound spectra of different household appliances with Sabine's annoyance contours, a contour for a satisfactory annoyance

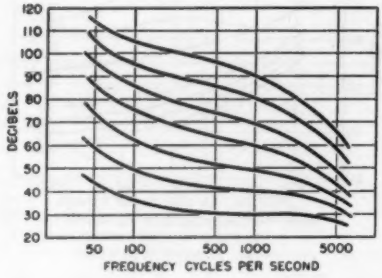

FIG. 2—EQUAL ANNOYANCE CONTOURS

level was taken as a goal. This contour was slightly higher but followed very closely the sound spectrum of a quiet oil burner.

Comparisons were also made to a third family of curves[3] showing the average reaction to different levels of outdoor noise in a residential area (see Fig. 3).

The least difference in sound intensity that the ear can detect is about 1 db. Doubling the intensity of a sound results in a 3 db increase, but doubling the loudness requires a 10-db or 10-fold increase in intensity; a 20-db increase or decrease signified a 100-fold change in intensity.

A diesel engine includes many possible noise sources. To reduce the noise to the established goal was a complex task. An accurate sound instrument was required that would separate and pin-point the source of each noise. For this purpose, an audio frequency spectrometer and automatic recorder was used. This instrument sweeps through the audio spectrum and draws it on a chart in 27 one-third octave steps with totals in both phons and decibels. The instrument can also be manually set for any particular frequency band. A probe microphone and a vibration pick-up were used with this instrument for hundreds of local and overall spectrum recordings that made a difficult task seem relatively simple.

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[1] Exponent numerals refer to the References.

SILENCING THE ENGINE

General Aspects: The diesel engine was silenced in a stepwise manner without any particular preconceived plan other than to attack the major noise makers first. As each major noise was reduced, other noises became apparent and were reduced in turn. Due to this method of approach, it is not possible to state just how much each component contributed to the total noise.

The diesel engine was first installed in an engine laboratory and bolted directly to the floor with the exhaust leading outdoors through a flexible pipe. The engine

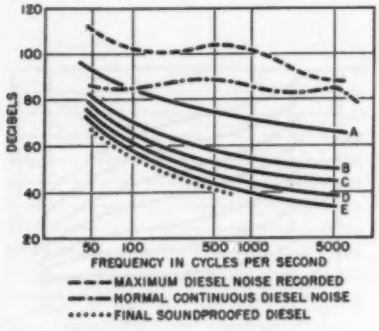

FIG. 3—A COMMUNITY'S REACTION TO OUTDOOR NOISE. CURVE A—VIGOROUS LEGAL ACTION; CURVE B—THREAT OF LEGAL ACTION; CURVE C—STRONG COMPLAINTS; CURVE D—MILD COMPLAINTS; Curve E—MILD ANNOYANCE

was operated running free. During starting and stopping cycles, sympathetic vibrations were generated, possibly at the natural frequency of the floor. When the engine settled down to its normal running speed, the sound level dropped to about 106 db.

Commercial silencers were then installed on the intake and exhaust and a preliminary study made of various types of engine supports. The engine was mounted on a heavy angle iron frame supported in turn on vibration eliminators, next on rubber engine mounts, and finally on a solid concrete block, weighing about 1000 lb and resting on another set of vibration eliminators. Spectra showed that the rubber engine supports reduced the total noise level by 2 db and 4 phons. The concrete block reduced the low end of the spectrum considerably; however, this region is of less importance than those at the middle and more ear-sensitive regions. For this reason, and since a concrete block would be costly, it was removed and the studies continued without it.

Supports and Vibrations: The diesel engine was next moved to a private laboratory and set up to drive the 5-hp heat pump. The engine was mounted on a steel

plate which was rigidly connected to the compressor mounting on the heat pump unit and a belt-driven counter shaft connected the diesel engine to the compressor. The steel plate supporting the engine was mounted on 4 commercial vibration isolators directly under the engine at one end, and the other end rested on the springs that had supported the 5-hp motor in the heat pump unit.

This arrangement resulted in severe vibrations and a careful study was made to determine the type and arrangement of supports that would transmit the least vibration. As a result, a final selection was made of rubber engine mounts directly under the engine plus vibration isolators with light springs. The cork snubbers in the isolators were replaced with butyl rubber which considerably reduced the horizontal vibration transmitted to the floor. The main purpose of the rubber engine mounts was to stop the mechanical transmissions of sound between the engine and supports. This combination of rubber and light springs resulted in almost com-

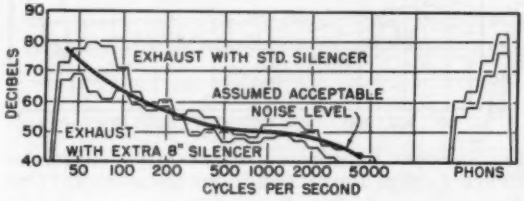

FIG. 4—EXHAUST SILENCING

plete elimination of the objectionable vibrations that had been originally transmitted to surrounding structures.

Combustion Air Intake: A soundproof box with walls 12 in. thick and designed for easy modification was next built around the engine, leaving the combustion air intake on the outside. Sound recordings showed a drop in the noise level across the entire spectrum by 5 db and 12 phons. However, the noise from the combustion air intake could now be heard over the total engine noise, showing that it had to be silenced too. For simplicity, the engine air intake was placed inside the soundproof box with air passing into the box through a labyrinth. This gave the desired reduction.

Engine Exhaust: Engine exhaust noise was studied next. As would be expected, the comparison showed the need for considerable noise reduction and a silencer was designed that would fit in the bottom of a chimney. This silencer was composed of an 8-in. diameter steel pipe with 4 solid baffle plates inside, spaced on 6-in. centers and connected by 1½-in. diameter nipples. Recordings made with this 8-in. pipe muffler showed the noise level to be below the assumed acceptable level right across the spectrum, as illustrated in Fig. 4. The exhaust pipe and silencers gave a total back pressure of 0.15 in. of mercury at 1200 rpm.

Miscellaneous Engine Noises: The noise level was now at a point where close attention had to be given to details. For this purpose a probe-microphone was used. The microphone picked up a noise from the fuel tank bracket that seemed to be as

loud as the rest of the engine. This bracket was bolted rigidly to the engine cylinder and was vibrating much like a loud-speaker. While a 6 × 3½-in. oval hole in the bracket eliminated most of this noise, the tank was removed, as it would be in a household unit, and noise level dropped 4 db.

Further studies with the probe-microphone to locate the source of peak noises that appeared in the overall spectrum led to minor changes that gave appreciable silencing. As typical of those changes, reductions of 6 db and 8 phons were gained by replacing the injection nozzle cover with a new, insulated cover. Noises transmitted mechanically along the fuel line to the soundproof box casing were eliminated by inserting a short section of rubber hose in the line. A 400 cps whine was traced to the air circulation fan blades on the fly wheel, there being 20 blades revolving at

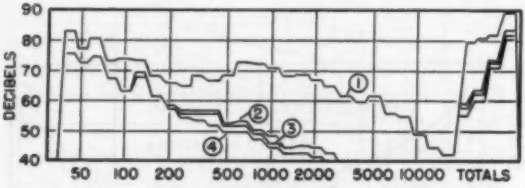

FIG. 5—SOUND REDUCTION FOR INSULATING MATERIALS. CURVE 1—BOX PANEL OPEN AND MICROPHONE 2 FT AWAY; CURVE 2—BOX PANEL CLOSED WITH GYPSUM BOARD; CURVE 3—PLEATED MINERAL WOOL ADDED; CURVE 4—¼-IN. WOOD FIBER HARDBOARD OUTER CASING ADDED

20 rps. This noise was cut down with a slotted metal cover to reduce impingement velocity of the entering air.

The miscellaneous engine noises had by now been so reduced that the entire character of the engine noise seemed changed. Instead of the original confusing clatter, the simple engine noises could now be clearly distinguished and the overall noise level of the engine open to the room was reduced from 106 to 90 db and from 114 to 80 phons. This is equivalent to a 15-*fold reduction* in loudness.

Sound Insulating Box: The engine noise was now at a point where further work on the engine seemed unnecessary, but the so-called *soundproof* enclosure was big and bulky. An investigation of different insulations was made in an effort to reduce its size. After studying many sound deadening materials in a multitude of combinations, the most effective was a 6-in. thick sandwich of a mineral-wool-faced gypsum board on the inside and a casing of ¼-in. tempered wood fiber hardboard on the outside with a pleated mineral wool blanket between. The progressive effects on the sound spectrum of adding these materials to the test panel are shown in Fig. 5. The total wall thickness might also be reduced from the present 6-in. thickness because, as the spectra show, nearly all of the sound attenuation is obtained by the gypsum board.

Spectra of the soundproofed engine alone, of the engine and compressor at different speeds, and of the complete diesel heat pump together with the assumed

acceptable noise levels are shown in Fig. 6. The curve of the assumed acceptable sound level is for conditions indoors. If the sound spectrum matches this curve 9 ft from the engine, then the noise would not be noticed through any floor or partition, because the average wall of a house will reduce the noise level by another 10 db or more, if there is no direct mechanical transmission.

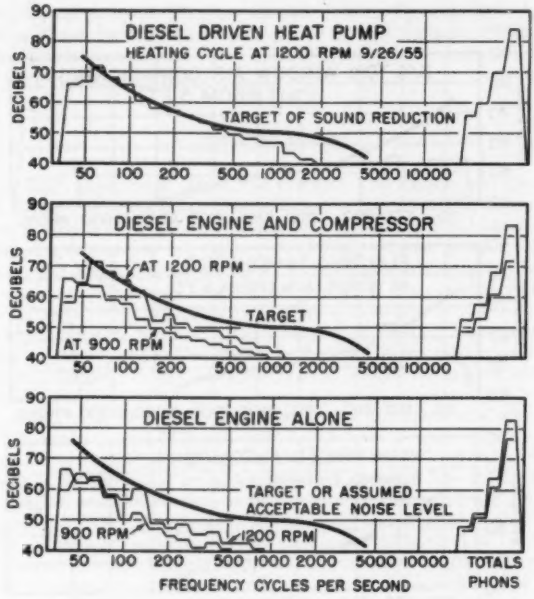

FIG. 6—FINAL SOUND SPECTRA OF DIESEL AND HEAT PUMP

For comparison of the soundproofed diesel engine with other noise producers, the spectra of a ¾-ton window air-conditioning unit, an electric typewriter at 65 words a minute, and a high-priced automobile idling, are shown in Fig. 7.

CONCLUSIONS

The conclusions gained from this study are that the noise of a diesel engine can be reduced to a satisfactory level for a heat pump drive, providing the entire unit is engineered and designed to take advantage of all the possibilities peculiar to this combination. The diesel engine is a reliable drive and maintenance should be reasonable, perhaps requiring a carbon clean-out about twice a year with a more com-

plete overhaul every 2 years. By using an enlarged oil pan, the oil could be made to last as long as a 275-gal tank of fuel and could be changed regularly by the fuel delivery man.

A well designed diesel-driven heat pump could be fully automatic, independent of outside power, and reasonably trouble free; it could fit inside a cabinet about 5 ft square × 4 ft high and could supply modulated year-round air conditioning at a lower operating cost and with less noise than the present all electric units. It

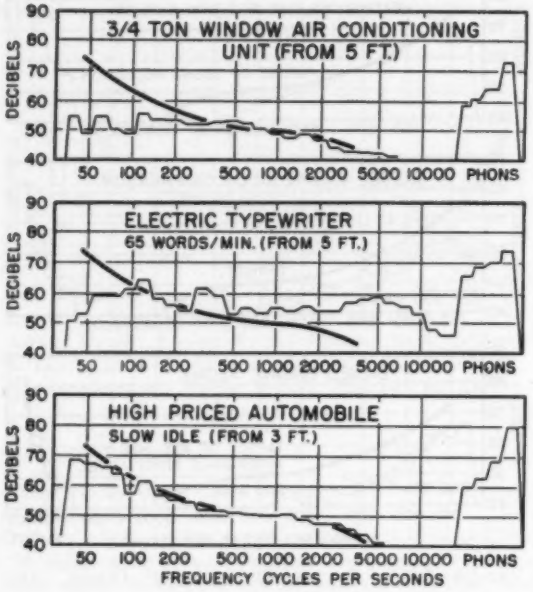

FIG. 7—SOUND SPECTRA OF OTHER EQUIPMENT COMPARED TO SPECIFIED TARGET

could also supply hot water and in addition, drive a generator for emergency home power.

A diesel-engine drive has, however, several disadvantages: (1) it will require a well-trained service man; (2) the compressor cannot be hermetically sealed, as are all the latest household refrigeration units; (3) the starting battery water and crankcase oil will need periodic inspection; (4) it requires a chimney for the exhaust; (5) it may have noticeable chimney exhaust odor under certain conditions; (6) it requires more floor space than other units of the same capacity.

Operating costs for the diesel, though not discussed herein, are lower than for the all electric unit. Investment costs are higher because of the higher first cost of the

diesel engine. The smaller size of the heat pump, when diesel driven, does save some investment but not nearly as much as the added cost of the engine. Also, while the cost of small diesels would decrease if mass produced for the heat pump market, the diesel driven unit would still cost more than the all electric unit. Rough estimates including maintenance show the operating savings might amortize the higher cost in 3 to 4 years in an average situation.

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1. Auditory Patterns, by H. Fletcher (*Review Modern Physics*, 1940, 47–65).
2. Control of Noise, by J. S. Parkinson (*Noise Control*, January 1955).
3. A Community's Reaction to Noise, by K. N. Stevens, W. A. Rosenblith and R. H. Bolt (*Noise Control*, January 1955).

DISCUSSION

J. B. CHADDOCK*, Cambridge, Mass., (WRITTEN): The following is a quote from the ASRE Data Book**: "It has been observed that a greater coefficient of performance could be obtained by driving the heat pump by means of internal combustion engines and recovering the heat from the cooling water and exhaust gases. Unique systems have been proposed and a number installed, however, for ordinary residential use this type of unit does not have a very great appeal in view of the customary greater attendance and maintenance required for internal combustion engines over the electric motor." The authors have conceded these disadvantages of a diesel driven heat pump, and have gone on to pioneer an investigation into another serious disadvantage, namely, the noise problem. I think they are to be commended on the results of that work which have been presented here.

The paper gives a value of 2.2 for the coefficient of performance of a heat pump operating between 0 F outdoors and 70 F indoors.

Let us assume that air will be delivered to the space at 110 F and that a 10 F temperature differential exists across the evaporator and condenser. For an ideal refrigerating cycle operating with a −10 F evaporator and a 120 F condenser then, the coefficient of performance is 3.4. Hence, the value of 2.2 used by the authors seems a conservative figure. The test values of coefficient of performance at the bottom of Table 1, however, seem to be in error. In the heating cycle test the outdoor temperature was 57 F, for which it would be expected that the CP would be about double the value at a 0 F outdoor temperature. In fact, using the 50,000 Btu per hr given in Table 1 as the sensible heat transfer to the air, and the work input as 4.0 hp, the CP may be computed as 50,000/(4)(2545) or 4.9. Similarly, I compute for the cooling cycle a CP of approximately 3.2.

As pointed out by the authors, a CP of 2.2 and an engine thermal efficiency of 28 percent gives a heat output of 2.2(0.28)(HV) = 0.615 HV, or 61.5 percent of the heating value of the fuel. This readily demonstrates the need for recovering the heat from the exhaust and from the cooling water jacket of the engine. The practical heat recovery from each of these latter sources would be about 30 percent of the heating value of the fuel. Presumably, the water circulated through the cooling jacket and exhaust jacket would be supplied at about 160 F to 180 F to a heat exchanger over which room air is circulated in winter and outside air in summer.

An important point which this paper brings up is the need for a reliable criterion for noise control. The authors have selected one of the *equal annoyance contour* curves of Hale Sabine as their criterion. The particular curve chosen being approximately equal

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* Massachusetts Institute of Technology.
** *Air Conditioning Refrigerating Data Book*, Design Volume, 1953–54, Page 18–20.

to that of a *quiet oil burner*. Chapter 41 on *Sound Control* in the 1957 Edition of the ASHAE GUIDE lists 4 commonly used methods of specifying noise. Briefly those are: (1) an octave band spectrum, (2) speech interference level (the arithmetic average of the 3 octave frequency bands from 600 to 4800 cps), (3) loudness level in phons computed from octave band levels, and (4) sound pressure level as measured on the A, B, or C scales of a sound level meter. The only one of these criterion which provides the essential information to a design engineer is the octave band analysis. In the case of this paper it is gratifying to find that a one-third octave band spectrum has been used.

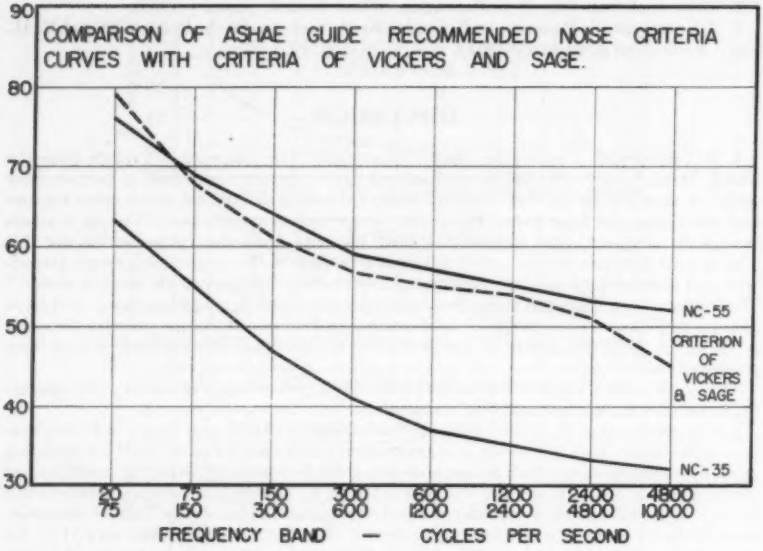

FIG. A-1—COMPARISON OF ASHAE GUIDE RECOMMENDED NOISE CRITERIA CURVES WITH CRITERIA OF VICKERS AND SAGE

Dr. Leo Beranek, who was responsible for the writing of the new chapter on *Sound Control*, has made several large scale studies of noise criteria in building spaces*. The results of these studies have been presented as Fig. 41.3 and Fig. 41.4 in the current edition of THE GUIDE. In Fig. A-1 the criterion used in this paper has been plotted in octave bands. The numbers were arrived at by adding 5 db to the values read from the *target of sound reduction* curve of the authors at the approximate geometric mean frequencies of each of the eight octave bands. This is a standard procedure for transposing third-octave band data to octave band data. Also shown in Fig. A-1 is the NC-55 criterion curve and NC-35 criterion curve of Dr. Beranek. The type of communication environment provided by these 2 latter curves is described as follows:

* Noise Criterion for Office Spaces, L. L. Beranek, Journal Acoustical Society of America, Vol. 28, September 1956.

NC–50 to NC–55, unsatisfactory for conferences of more than 2 or 3 people: telephone use slightly difficult; normal voice intelligibility from 1 to 2 feet; raised voice 3 to 6 feet.

NC–30 to NC–35, *quiet* office, satisfactory for conferences at a 15 foot table; normal voice intelligibility 10 to 30 ft, telephone use satisfactory.

A desirable noise level for home sleeping areas has been given as NC–25 to NC–35**. Although such levels are commonly exceeded in many homes today, I believe it should be the goal of air conditioning and acoustical engineers to design for these levels.

The manner in which the authors have systematically approached the noise reduction problem of their diesel engine should be made required reading for the designers of air-conditioning equipment.

It is interesting to note how closely the ordinary *mass law* predicts the sound transmission loss through the mineral-wool-faced gypsum board enclosure. For a single damped wall, homogeneous in structure, the transmission loss (TL) is dependent primarily on the product of the surface density (pounds per square foot) and the frequency. Assuming a density of 1 lb per sq ft for the gypsum board, I have determined the TL by mass law and compared it with the values measured by the authors as shown in their Fig. 5. The results are shown in Table 1–A.

TABLE A-1—TRANSMISSION LOSS (TL) AS CALCULATED BY MASS LAW AND AS FOUND BY AUTHORS' TESTS

FREQUENCY CPS	TL BY MASS LAW, IN DB	MEASURED TL, CURVE 1 — CURVE 2 OF FIG. 5, IN DB
100	7	7
500	17	15
1000	23	22
3000	32	25

AUTHORS' CLOSURE (R. W. Sage): I would like to thank Professor Chaddock for his comments on our paper. I think I should first apologize for the coefficient of performance data that are included in the preprints. As Professor Chaddock has pointed out they do not agree with the usual COP definition. We essentially agree with the numbers he has proposed in their place, namely, a 3.2 COP for the cooling cycle and a 5.6 COP for the heating cycle. The numbers in the paper, 1.01 and 1.46 for the two cycles, respectively, are based on overall oil efficiency. To those of us in the oil industry, this is a measure of how effectively our fuel is performing. This change is incorporated in the paper as it appears in the TRANSACTIONS.

Professor Chaddock's comments on the criteria employed for *a satisfactory sound level* are interesting and informative. In general the NC–35 level he indicates as satisfactory closely approaches our goal when including the additional attenuation through residential floors and walls as pointed out in the paper. This would vary from home to home and in all instances care would have to be taken to be sure that a diesel driven unit was not located directly beneath the sleeping quarters or where the floor had unusually poor attenuation characteristics. Also, in an individual installation, the home owner must be taken into consideration. In another study we found that with identical noise levels at the source in 2 different homes, a considerable variation in annoyance occurred. While this was partially a function of home construction, perhaps even more important, it was caused by a different sensitivity of the occupants to noise.

It was interesting that the calculations by Professor Chaddock using the *Mass law* predict so closely the sound transmission loss through the gypsum board enclosure.

** Table 41.4, HEATING VENTILATING AIR-CONDITIONING GUIDE 1957, Chapter 41.

I would like to close my comments by emphasizing what we feel is a basic conclusion reached from a study of this type. It is this: noise making equipment can be silenced to extremely low levels by proper attention to detail and application of basic principles for analyzing and removing sources of sound. To us it was surprising that the results achieved with the diesel engine could be obtained with an engine that is so seriously noisy in our every day experiences. The silencing here was achieved not by a single magic touch applied to the overall engine, but rather by the patient tracking down of each individual noise maker associated with the machine. Although the contribution of each might be judged small, nevertheless, by eliminating the small noises, the overall noise level was drastically reduced. In this everyday world where noise seems to be continually on the increase, this lesson seems to take on added importance. Mechanical equipment of all types and particularly in the heating and air-conditioning field could be substantially quieter. Frequently these improvements would not cost additional money. The result would be a quieter America and I think we would all agree that this is desirable. .

I would like to thank all who have commented. The authors enjoyed the opportunity of making this presentation to the Society and hope to have similar subjects for presentation at future meetings.

1612

No. 1612

ACOUSTIC COUPLING OF RESIDENTIAL FURNACES WITH THEIR SURROUNDINGS

By C. F. SPEICH*, W. R. DENNIS**, AND A. A. PUTNAM***, COLUMBUS, OHIO

This paper is the result of research sponsored jointly by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, the *American Gas Association* and the *Oil-Heat Institute of America, Inc.*, conducted through Battelle Memorial Institute.

PULSATIONS have been one of the most annoying problems facing the manufacturer and the installer of combustion equipment for heating. A furnace or boiler that *goes into pulsation* may be noisy enough to be heard or felt throughout the dwelling, and sometimes may even cause damage to the building or its contents. Service calls to correct the condition can represent a considerable expense to the organization responsible for the satisfactory performance of the equipment. Therefore, a research project was activated at Battelle in May 1954, to (1) determine the nature of pulsations in oil- and gas-fired residential furnaces and boilers, and (2) find methods for suppressing pulsations. This work has been made possible by the cooperative efforts of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, the *American Gas Association*, and the *Oil-Heat Institute of America, Inc.*; support within the latter organization has come both from the equipment manufacturers and from oil refiners. The project is guided by the Pulsations Research Steering Committee of the Technical Advisory Committee on Combustion of the ASHAE.

The purpose of this paper is to show that the room in which the heating unit is located can play, on occasion, an important role in the overall room noise level produced by the combustion process in the heating unit. Although this paper deals specifically with residential gas- and oil-fired furnace-burner units, it should be emphasized that the methods outlined here are equally applicable to boilers and other heating units. The important criterion in determining if a unit will be troublesome is not its size, but instead, its frequency of oscillation relative to the natural or resonant frequencies of the enclosure in which it is installed.

It should also be stated that the noise levels referred to in the text are those at different points in the room which result from the combustion noise originating

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Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Murray Bay, Que., Canada, June 1957.

TERMINOLOGY

Before discussing the theoretical considerations and the results of the tests, a number of acoustical terms will be defined.

Fig. 1 shows the sinusoidal variations of pressure with time and those terms which may be clarified by illustration are shown on the figure.

1. *Oscillation* — periodic variation, with time, of pressure with respect to the ambient pressure. During oscillation, the pressure is alternately greater than and less than the ambient pressure.

2. *Pulsation* — here considered to be the observable effects of the pressure oscillations. That is, pulsations can be seen, felt, or heard by an observer, or measured as changes in pressure of the surrounding atmosphere or as forces exerted on surrounding surfaces. Since there are certain physiological implications connected with the term *pulsation*, the term *oscillation* is used in the subsequent discussion.

3. *Cycle* — complete sequence of values of pressure; at the end of a cycle, the pressure, the rate of change of pressure, and the direction of change of pressure have the same values that they did at the start.

4. *Period* — time interval consumed by one cycle.

5. *Frequency* — number of periods occurring in 1 sec, or number of cycles per second (cps).

6. *Amplitude* — absolute value of the maximum difference between the instantaneous static pressure and the ambient pressure in a given period. The variations in the instantaneous static pressure can be with respect to either time or space and the value of the amplitude need not be the same for both.

7. *Phase* — fractional part of a period through which the independent variable, usually time, has advanced, measured from an arbitrary origin. The origin is usually chosen so that the fraction is less than one.

8. *Wavelength* — distance between 2 wave fronts in which the pressures have a difference in phase of 1 complete period. During one period, the wave has advanced a distance equal to the product of the velocity of sound and the period.

9. *Standing Wave* — oscillations having a fixed distribution of amplitudes in space, and characterized by the existance of nodes and antinodes that are fixed in space.

10. *Nodes* — points, lines, or surfaces in a standing wave where the instantaneous pressure is equal to the ambient pressure during a given period.

11. *Antinodes* — points, lines, or surfaces in a standing wave where the instantaneous static pressure is equal to the amplitude during a given period.

12. *Natural Frequency* — the frequency at which a system will tend to oscillate when disturbed. The natural acoustic frequency of an enclosed space is a function of the configuration of the space.

13. *Resonance* — for this discussion, resonance can be considered to be a condition in a system where the frequency of a periodic, driving force on the system is equal to the natural frequency of the system. The effect of resonance is to cause oscillations of usually large magnitude in the system and to allow coupling for maximum energy transfer to take place between the system and the driving force.

14. *Resonant Frequency* — the frequency of the periodic driving force which causes a system to resonate.

15. *Mode Number* — a mode number is equal to the number of half wavelengths along a given dimension (in rectangular coordinates) for a given system.

within the combustion chamber of the burner units. The term *coupling* used in the title and text normally might imply that the noise from the combustion chamber affects the noise level in the surrounding room, and that the reverse is also true. However, in this study, the reverse effect appears to be quite small. For as outlined in an earlier paper,[1] the mechanisms that generate the oscillations in gas- and oil-fired furnace-burner units are involved with quite local conditions within the furnace proper. Changes in the pressure within the furnace resulting from the oscillations in the surrounding room are felt to have only a slight effect on

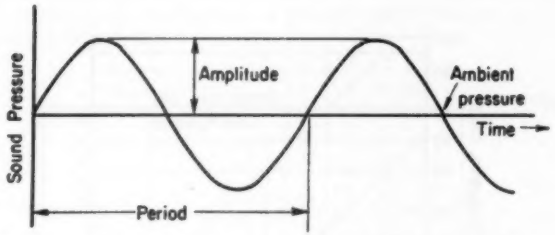


FIG. 1—SINUSOIDAL VARIATION OF PRESSURE WITH TIME

those local conditions which cause furnace oscillation. Thus, the term *coupling* in this paper should be thought of in a more restricted sense than usual.

PRELIMINARY CONSIDERATIONS

A series of field tests was made to determine if there was a correlation between the oscillating frequency of different residential furnaces and the distance between the entrance of the air into the furnace and the barometric damper for oil furnaces or the draft hood for single-port-type gas furnaces.

Fig. 2 shows a log-log plot of the results of these field tests. It should be noted that these oscillating frequencies were fixed by design and were not varied during a particular test. The method of approximating the distance of gas travel inside the furnace is outlined in Fig. 7 of the earlier paper.[1] On Fig. 2, straight lines at an angle of 45 deg are drawn through each set of data.

If all the furnaces tested had been identical except for size, then theoretically the points would have fallen on such 45 deg straight lines as shown on Fig. 2. It is evident from the scatter of the data that the furnaces were not identical. Nevertheless, by using these straight lines, a first approximation can be obtained for the value of the frequency of oscillation for a particular furnace. It should be noted that all the furnaces listed had frequencies of oscillation below 80 cps.

Any room or large enclosure will have several resonant frequencies and these frequencies will be a function of the dimensions of the room. Generally speaking, rooms usually encountered in residential installations will have at least one resonant frequency between 10 and 100 cps. If a furnace which oscillates at a certain frequency is put in a room whose resonant frequency is close to the oscillating fre-

[1] ASHAE RESEARCH REPORT NO. 1596—Pulsations in Residential Heating Equipment—Preliminary Results, by A. A. Putnam and W. R. Dennis (ASHAE TRANSACTIONS, Vol. 63, 1957, p. 153).

quency of the furnace, then there may be a general increase in the noise level produced by the furnace. Essentially, the room amplifies the noise of the furnace; in the technical sense there is an *acoustical coupling* between the furnace and the room. A standing wave is set up and in certain parts of the room the noise level will seem higher than normal.

EXPERIMENTAL RESULTS

Fig. 3 is a plan view of the room chosen for the series of tests designed to show that acoustic coupling can occur between a furnace and a room. The room was

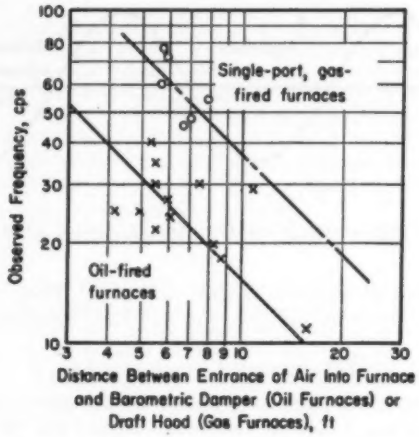


FIG. 2—FREQUENCY OF OSCILLATION AS A FUNCTION OF FURNACE SIZE

24 ft 7 in. long, 11 ft 6 in. wide, and had a height of 10 ft-10 in. to the bottom of the ceiling joists. Although there was a jog in wall A, it will be shown later that this had no appreciable effect on the results of the tests.

In the first series of tests, a loudspeaker, mounted in a cabinet, was used as the sound source. The speaker was driven by a audio oscillator. The "X" marked on the centerline of the room, as shown in Fig. 3, indicates the position of the condenser microphone used in these tests. The signal from the microphone was fed into a chart recorder which recorded both the magnitude and frequency of the signal.

The first test conducted was to determine the frequency response of the room and, in particular, the resonant frequency of the room. This was done by placing the loudspeaker-cabinet combination on the centerline of the room near wall B. The frequency of the sound was varied and the amplitude of the sound level recorded for each frequency.

Fig. 4 shows the results of these tests. The resonant frequency occurred at a frequency of 26.5 cps. In order to assure that this resonant point was due to the room and not the loudspeaker-cabinet combination, a test was conducted on the

loudspeaker-cabinet combination alone in a large room. The results of this second test are shown in the inset in Fig. 4. As can be seen, there was no resonant frequency of the loudspeaker-cabinet combination between 19.5 and 34.5 cps. Therefore, it can be concluded that the resonant frequency of the room was 26.5 cps.

In the second series of tests, the microphone was kept in a fixed position, but the speaker-cabinet combination was moved along the centerline in such a way as to be always parallel to the original position. A constant signal, whose frequency was 26 cps, was fed into the loudspeaker.

Fig. 5 shows the average amplitude of the sound at the microphone as a function of speaker position. As can be seen, a standing wave was set up in the room. There appears to be a pressure antinode at or near wall B.

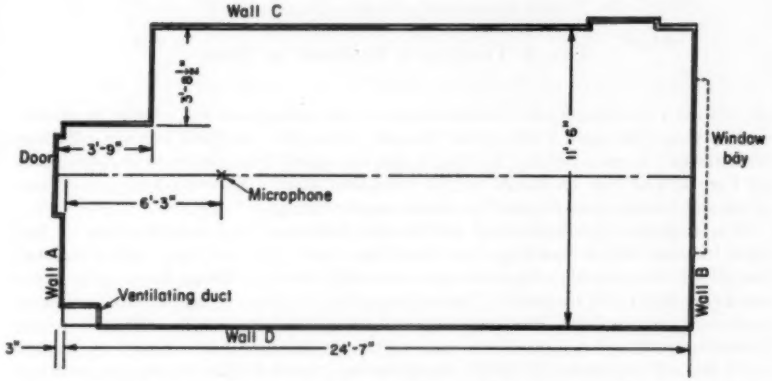
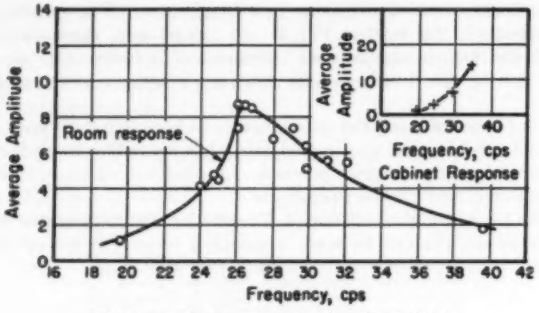


FIG. 3—FLOOR PLAN OF ROOM USED FOR RESONANCE TESTS

Fig. 6*a* shows the results of a test in which a residential oil-fired furnace, whose oscillating frequency was about 25 cps, was substituted for the loudspeaker-cabinet combination. This particular oil furnace was used because it had an oscillating frequency close to the resonant frequency of the available room. This oscillating frequency was kept constant during the test by maintaining the furnace controls, the distance from the furnace to the barometric damper, and the overall stack length constant. It can be seen that a curve similar to that of Fig. 5 was obtained.

Fig. 6*b* indicates that the average amplitude of oscillation in the combustion chamber of the furnace did vary somewhat during the test. Correcting the curve in Fig. 6*a* for the changes in combustion chamber noise would have the effect of altering the position of the curve, but would not greatly change the shape of the curve. As a result, the change that did occur in the amplitude of the combustion chamber noise can be neglected.

Fig. 6*a* also shows the ability of a room to amplify the noise of a furnace. When the furnace was located near the center of the room, the noise level at the microphone position was quite low. However, as the furnace was moved closer to wall B, the noise level at the microphone increased greatly. Because of the charac-

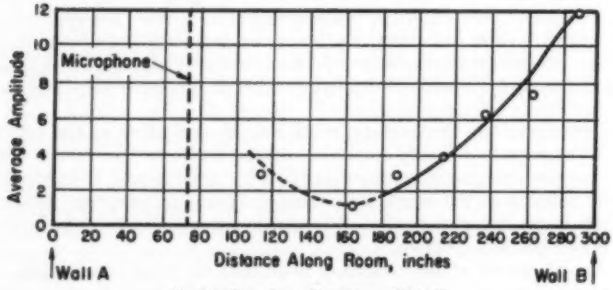


(Inset shows frequency response of speaker cabinet alone. Microphone at position marked in Fig. 3)

FIG. 4—FREQUENCY RESPONSE OF ROOM

teristics of a standing wave, the noise level at the microphone will usually be greater or less than the noise level at the furnace, depending on their relative positions with respect to each other. It should also be added that the vertical coordinates on Fig. 6*a* and Fig. 6*b* should not be compared since 2 different microphone systems and positions were used for these measurements.

If a residential gas furnace, of similar size, had been used instead of the oil furnace for this test, a standing wave could also have been set up. Research work has shown that single-port gas furnaces normally have oscillating frequencies which are twice that of oil furnaces. Since the product of the wavelength and frequency is a constant, doubling the frequency will halve the wavelength. Therefore, one complete cycle will appear in the case of the gas furnace as compared to one-half cycle for oil furnaces. It would therefore be expected that a pressure antinode would appear in the center of the room as well as at each end. For reasons discussed later, the locations for the gas furnace resulting in a minimum coupling with the room and therefore minimum noise output, would have been one-quarter of the room length from either end.



(Outlet of speaker cabinet facing Wall B. Microphone at position marked in Fig. 3)

FIG. 5—EFFECT OF SPEAKER CABINET POSITION ON AMPLITUDE

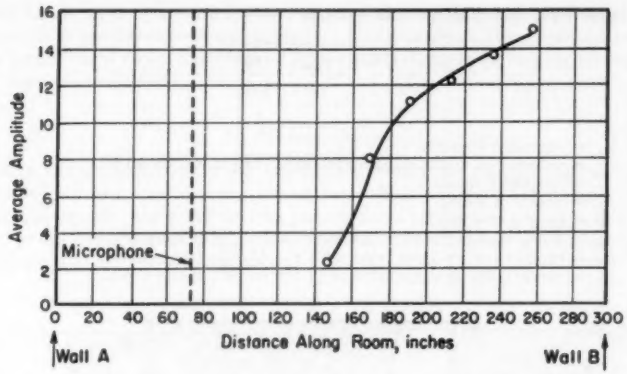


FIG. 6a—EFFECT OF OIL FURNACE POSITION ON AMPLITUDE

THEORETICAL CONSIDERATIONS

The results of the test described show that coupling can take place between a furnace and a room and therefore indicate the desirability of preventing such coupling. Fig. 6*a* shows the importance of the proper furnace location. By positioning the furnace near the pressure node, coupling is prevented. When room dimensions can be changed, it is possible to make a judicious choice of room dimensions that will prevent coupling. In either case, it is important to know the frequencies of the modes of oscillation that can exist in a given room so that they can be compared with the frequency of oscillation of the furnace in question.

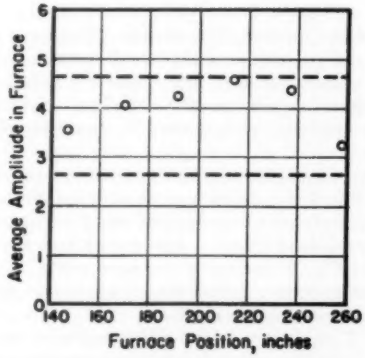


FIG. 6b—AMPLITUDE IN COMBUSTION CHAMBER (PROBE MEASUREMENT)

The basic equation used in this work is

$$f_n = \frac{c}{2}\sqrt{\left(\frac{n_l}{l}\right)^2 + \left(\frac{n_w}{w}\right)^2 + \left(\frac{n_h}{h}\right)^2} \quad \text{.....(1)}$$

where

f_n = resonant frequency.
c = velocity of sound (approximately 1120 ft/sec at 60 F).
l = length of room.
w = width of room.
h = height of room.
n_l = mode number associated with the length of room.
n_w = mode number associated with the width of the room.
n_h = mode number associated with the height of the room.

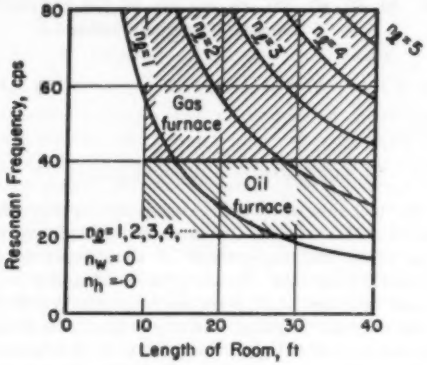


FIG. 7—THE RESONANT FREQUENCY OF A ROOM AS A FUNCTION OF THE LENGTH OF THE ROOM

Equation 1 is used to determine the resonant frequency in terms of the 3 dimensions of a room and the mode numbers. A mode number can only have integral values greater than or equal to zero. For any particular natural frequency of the room, each dimension will have an individual mode number which will be equal to the number of evenly spaced lines of zero pressure amplitude (node) in that direction.

Figs. 7, 8, and 9 represent plots of this equation for one, two, and three dimensions, respectively. Fig. 7, for which n_w and n_h equal zero, indicates the resonant frequency as a function of mode number and one dimension. Fig. 8 indicates the resonant frequency as a function of 2 dimensions and the mode numbers $n_l = n_w = 1$, $n_h = 0$. In Fig. 9, the mode numbers are $n_l = n_w = n_h = 1$, and w is also fixed at a length of 20 ft. For all these plots, those areas which are pertinent to oil furnaces and gas furnaces are shaded.

Figs. 10, 11, and 12 are examples of 3 different modes of oscillation that might be commonly found in rooms. The shaded areas are indicative of the positions of the pressure antinodes for standing waves which might exist. These areas, therefore, represent unfavorable positions for a furnace. As an example, Fig. 10 cor-

responds to the case of the oil furnace already discussed and Fig. 11 corresponds to the case of a gas furnace of similar size. An example of Fig. 12 would be an oil furnace located in a room the width of which was too large to neglect in determining the resonant frequency.

APPLICATION

By using the equation and plots which have been presented, it is often possible to predict the proper locations for oil and gas furnaces to give low noise level. In applying this theory, there are certain assumptions which can be made depending on the type of furnace used and the size of the room. Because of this, the oil and gas furnaces will be treated separately.

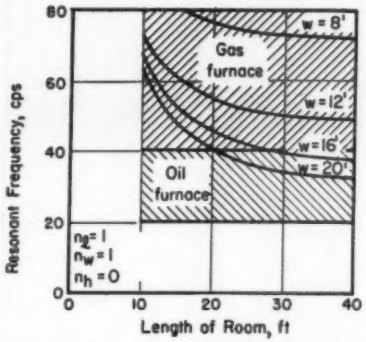


FIG. 8—THE RESONANT FREQUENCY OF A ROOM AS A FUNCTION OF THE LENGTH AND WIDTH OF THE ROOM

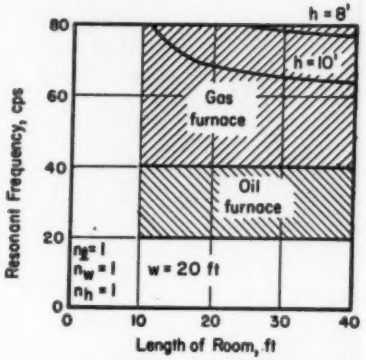


FIG. 9—THE RESONANT FREQUENCY OF A ROOM AS A FUNCTION OF THE LENGTH, WIDTH, AND HEIGHT OF THE ROOM

Oil Furnace: For an average size of room, the height is of no consequence in the determination of the number of modes of oscillation. Also, for rooms up to 40 ft long, the width need not be considered if it is under about 15 ft. Under these circumstances, Fig. 7 can be used. The figure shows that only 2 modes, the first and second, can exist in the range of frequencies of interest. For rooms between 14 and 28 ft in length, the first mode is possible. If the frequency of oscillation of the furnace is suspected to be such as to give coupling, the furnace should be equally spaced as nearly as possible between the 2 ends. For larger rooms, the second mode might be present which would cause a pressure antinode to appear in the center of the room as well as the ends. For this situation, the furnace might be placed at the quarter points.

For rooms wider than 15 ft, both Figs. 7 and 8 must be used in order to predict coupling. If a standing wave of this mode type is set up in the room, the pressure antinodes will appear in the 4 corners, as indicated in Fig. 12.

As a first example, consider the room which was used for the series of tests which have been described. As was previously mentioned, this room was about 25 ft long, 12 ft wide and 11 ft high. Since an oil furnace was used for these tests, the

oscillating frequency would have been between 20 and 40 cps. Fig. 9 indicates that a room 11 ft high could not have a combined resonant frequency in this range. Therefore, the height of the room is of no consequence. Likewise, Fig. 8 shows that a room 12 ft wide would not have a combined resonant frequency below 40 cps. Therefore, the width of the room can also be neglected.

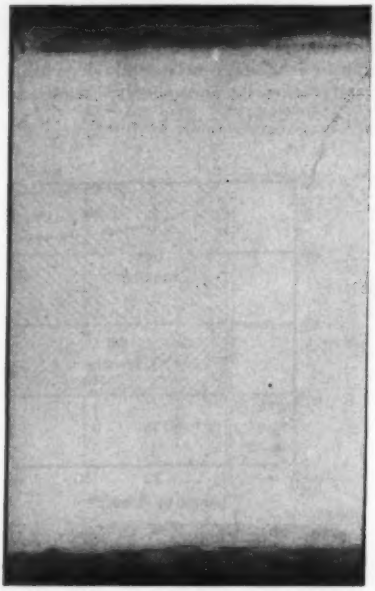


FIG. 10—SKETCH OF POSITIONS OF PRESSURE ANTINODES FOR ONE DIMENSION AND THE FIRST MODE OF OSCILLATION

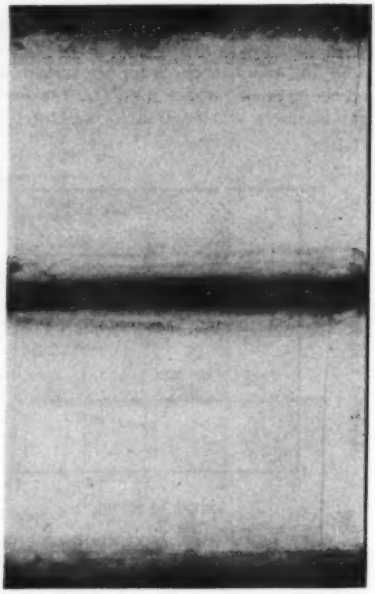


FIG. 11—SKETCH OF POSITIONS OF PRESSURE ANTINODES FOR ONE DIMENSION AND THE SECOND MODE OF OSCILLATION

Using Fig. 7, it can be seen that a length of 25 ft will give a single mode of oscillation whose frequency is 21 cps. This is in reasonably close agreement with the observed value of 26 cps. The most probable reason for the discrepancy is the jog in wall A of the room, which shortens the wavelength of the sound wave somewhat.

As another example of the use of Figs. 7, 8, and 9 suppose an oil furnace, which was known to oscillate at a frequency of 30 cps, were to be placed in a room where dimensions are 30 ft long, 10 ft wide, and 8 ft high. Figs. 8 and 9 indicate that neither the width nor height of this room needs to be considered for a furnace oscillating at a frequency of 30 cps. Fig. 7 shows that there is no mode of oscillation close to 30 cps that can exist in a room 30 ft long; hence, no coupling should exist between the furnace and the room. If instead, the room had been 37 ft long, with the other dimensions the same, then trouble with the second mode of oscillation could be expected and the furnace should either be located near one of the quarter points, or a furnace of different oscillating frequency used. It should

be added that a tolerance of at least ±4 ft is recommended when considering dimensions to prevent coupling. This is because of the wide peak in the frequency response curve around the resonant frequency, as shown in Fig. 4.

Gas Furnace: Because of the higher frequencies involved with gas furnaces, more possible modes of oscillation usually have to be considered than with oil furnaces.

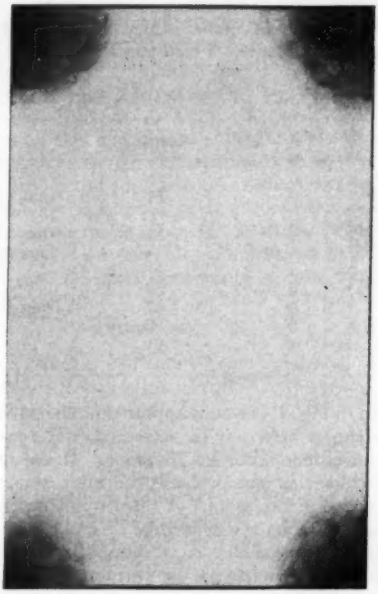


FIG. 12—SKETCH OF POSITIONS OF PRESSURE ANTINODES FOR THE FIRST MODES IN TWO DIMENSIONS

As a result, the analytical problem of determining the possibilities of coupling becomes somewhat more difficult. Since plots cannot adequately represent all the dimensional possibilities that can exist, the importance of the equation for determining room resonant frequencies cannot be overly emphasized.

For instance, Fig. 8 shows the modes $n_l = 1$, $n_w = 1$, $n_h = 0$; the modes $n_l = 2$, $n_w = 1$, $n_h = 0$ or $n_l = 1$, $n_w = 2$, $n_h = 0$, which would also be of interest in the case of a gas furnace, would require additional plots.

As an example of the analytical procedure to be followed for a gas furnace, consider the 30-ft by 10-ft by 8-ft room discussed with reference to the oil furnace. Using Equation 1 and the first (1) modes of oscillation in 3 dimensions, the dimensions will give a resonant frequency of 91 cps. Because this frequency is greater than 80 cps, it is possible to have cases where the height of the room can be neglected; thus $n_h = 0$. If $n_h = 0$, the resonant frequency of the room will be lower and within the 40 to 80 cps range as is shown in Table 1.

It would therefore appear that a gas furnace with an oscillating frequency below about 52 cps would cause no coupling difficulty in the room at any location. On the other hand, for a furnace with an oscillating frequency of about 57 cps, the furnace should not be located at (*a*) the ends of the room or one-third the length from either end, (*b*) the long walls of the room, or (*c*) in the corners. These 3 results come from consideration of the regions of maximum pressure amplitude for the 3 different modes of oscillation at about 57 cps.

Rooms Other Than Rectangular in Shape: Many times, the room under consideration will not be rectangular in shape. From consideration of the results obtained

TABLE 1—RESONANT FREQUENCY OF 30- x 10- x 8-FT ROOM FOR GAS FURNACE

n_l	n_w	n_h	FREQUENCY	REMARKS
3	0	0	56	
4	0	0	74	from Fig. 7
0	1	0	56	from Fig. 7 using dimension of 10 ft
1	1	0	59	
2	1	0	67	from Equation 1
3	1	0	80	
1	1	1	91	

in the test room, shown in Fig. 4, it would appear that the deviations from the ideal shape of the rectangle shown here may be neglected. The problem becomes more difficult when the room is more complex in shape. If the room, for instance, is L-shaped, 2 of the frequencies can probably be approximated quite closely by using the length and width of the room. A third frequency will probably be consistent with a mean length along the centerline of the 2 arms of the room.

Calculation of all the possible modes of oscillation in these cases is impractical. The only satisfactory way to determine the various modes, frequencies, and corresponding points of maximum pressure amplitude is by use of a sound generator in the actual room to determine its resonant frequency, as was described in this report. It is also possible to get this information using a scale model of the room providing that the model is geometrically similar to the actual room. It should be added that the frequencies obtained in this manner will be proportional to the actual frequencies; the proportionality constant is the inverse of the scale factor of the model.

Determining the Natural Frequency of a Furnace: If it is desired to determine the natural frequency of a furnace to a closer value than the range of scatter that is indicated by Fig. 2, then the most economical means is probably the use of electronic sound-measuring equipment in the manner similar to that discussed in the earlier paper.[1] On the other hand, it is feasible to install the furnace at one end of a solidly built corridor, and move a heavy wall in from the other end. Having determined the length of the corridor for which the amplitude appeared largest to the ear, the natural frequency could be determined from Fig. 7. The movable wall, however, would have to be of concrete block or similar construction, probably mounted temporarily on a lift truck. The fit to the edges of the corridor at each test point would have to be such that the gap did not exceed one quarter to one-half inch.

CONCLUDING REMARKS

It has been shown that coupling can occur between a residential furnace-burner unit and the room in which it is placed. When such a condition exists, a normally quiet unit may be entirely unsatisfactory for the given installation. Once the furnace-burner unit is installed, it is often impractical to relocate the unit to give lower noise levels. In situations of this nature, it might be possible to build a partition in the room which will break up the standing waves. Such a wall could have the additional advantage to the consumer of concealing the equipment.

A more direct application of this paper may be found for those people who install units large enough to justify the cost of investigating possible positions of the unit to give low noise levels before the units are installed.

Another application of this material would be as another tool in *trouble-shooting*. Considerable time might be saved with unusually noisy units if the possibility of coupling between the room and furnace were checked out early in the investigation of the noise problem. One symptom which would be an indication of coupling would be the appearance in some areas of the room, possibly quite far from the furnace, where the noise level would be high, whereas in other positions, possibly quite close to the furnace, the noise level would be considerably lower.

ACKNOWLEDGMENT

The able assistance of the members of the Pulsation Research Steering Subcommittee of the TAC on Combustion of ASHAE in guiding the research and providing necessary information and materials is gratefully acknowledged.

DISCUSSION

W. W. POPYK, Johnstown, Penna. (WRITTEN): The paper and the earlier one, *Pulsations in Residential Heating Equipment* have been of considerable interest to many involved in the manufacture, installation and servicing of residential heating equipment.

These papers have conveyed a very accurate analysis of the source and cause for amplification of the resulting noises of combustion. These were based on a scientific investigation which should eventually lead to practical design, installation and servicing measures.

The present paper is based on the fact that all appliances generate combustion noises regardless of size and the criterion in determining whether the unit will be troublesome is its frequency of oscillation relative to the natural enclosure in which it is installed.

I cannot help but recall some of our noise difficulties and experiences which would warrant consideration in a brief discussion of this subject, viz.-

1. Most of our noise complaints were confined to one particular type of heating boiler design, consisting of several sizes ranging in oil firing rates from 0.85 to 1.65 gal per hr.

2. Noises were detected in different and varying areas of the building regardless of whether the heating unit was in an open basement or closet isolated installation.

3. One particular heating season evidenced more difficulty of this type and greater severity than any other period before or since.

4. Though not conclusive, much relief from noises was obtained by modifying the oil burner blast tube to one having less resistance to air flow delivery.

These facts with other related observations would further indicate that:

1. Oscillating qualities must differ or vary with different types and sizes of heating units.

2. Fuel variations apparently have a very marked effect on combustion quality.

3. Preparation and delivery of fuel and air will affect both combustion and circulation of products of combustion in a heating unit and result in varying degrees of oscillation.

Therefore, we would not question that buildings and surroundings can be associated with combustion acoustics and can be controlled within acceptable limits as concluded in this paper, but obviously, something is occurring in the combustion process which cannot be accepted as a criterion for change in building construction and configuration to overcome noise.

The type of noise difficulty which we are concerned with came into existence in recent years with the catalytically blended fuels and the smaller and tighter building construction. Heating loads have been decreased accordingly and heating units have been designed respectively for compactness and higher operating efficiencies.

However, with the known variations in fuel constituents and combustion characteristics, there have been no significant changes in design of firing devices. This would appear to be the more logical phase for continued investigation to pursue.

W. T. KNOX, Linden N. J.: If I understand correctly the prepared comments, there were some implications that the fuel composition and combustion characteristics might play an important part in the noise generated by the furnace. I do not for a moment pretend that we have made any exhaustive study of the problem of noise versus fuel quality, but we have studied for many years the combustion of different fuels falling within *ASTM* No. 1 and No. 2 heating oil grades, in domestic boilers and furnaces. These fuel oils have varied from being very aromatic to very paraffinic. So far as we have been able to determine, the combustion characteristics (with gun type burners where the mixing of air and fuel spray was good) of these varying kinds of oil have been quite similar. Judging by length of flame and other visible characteristics of the flame, we have been unable to discover any significant difference in combustion characteristics with these rather widely varying fuel qualities. We question, therefore, the suggested relation between noise and fuel composition.

AUTHORS' CLOSURE (Mr. Speich): As Mr. Popyk states, the paper points out that all heating units emit combustion noise to some extent and that this combustion noise need not be objectionable. However, as the paper seeks to show, the location of the furnace-burner unit in a room as well as the size of the room can cause the room to amplify combustion and pulsation noise from the furnace. This effect can take place because of an acoustic coupling between the frequency of the combustion noise from the furnace-burner unit and the resonant frequencies of the room. The paper also points out that this amplification problem can be alleviated by the proper positioning of the furnace-burner unit in the room or by a judicious choice of room size.

Although the paper deals specifically with the effect of a room on the noise output of a furnace-burner unit, it should be realized that the room-furnace relation is only one of several criteria which can produce a noisy installation. The furnace-burner unit itself can produce objectionable noise without room acoustics entering the picture. Combustion noise from furnace-burner units alone was the subject of a previous paper entitled ASHAE RESEARCH REPORT No. 1596—Pulsations in Residential Heating Equipment — Preliminary Results, by A. A. Putnam and W. R. Dennis (ASHAE TRANSACTIONS, Vol. 63, 1957, p. 153).

The observations which Mr. Popyk has made are understandable, and the basic reasons which lie behind these observations are the subject matter of the present research program. Our research on oil-fired units, for instance, has shown that there is a great interdependence among the fuel spray, air, and recirculation patterns in and around the flame, and their combined effect on noise. It is the relation of each of the 3 with each other which appears to govern whether pulsation will occur or not, and to what degree of amplitude it will occur.

Since we have not had the opportunity to investigate the effect of fuel composition on the amplitude of oscillation in oil furnaces, I want to thank Mr. Knox for his additions to this discussion.

1613

No. 1613

PERFORMANCE OF INTERMITTENTLY-FIRED OIL FURNACES

By W. G. COLBORNE*, KINGSTON, ONT., CANADA

OUTPUT of an oil-fired furnace is normally controlled by a thermostat which starts and stops the burner. For reduced outputs the burner operates for shorter on-periods relative to the off-periods. The actual length of the cycle will vary between different installations even for the same average hourly heat output.

In order to predict how a furnace will operate in an actual installation, a test which is run under intermittent-firing conditions will be more valuable than a test which is run under steady-state conditions. The purpose of this project is to investigate the performance of 2 standard warm air furnaces under various conditions of intermittent firing.

APPARATUS AND TESTS

The laboratory arrangement of ductwork, furnace and measuring equipment is shown schematically in Fig. 1. The Pitot tube and slope gauge were calibrated at the center of the throat section of the venturi. The calibration was made from a 20-point traverse, using a micromanometer for pressure measurements. The calibration curve was a plot of the average velocity at the 20 points against the velocity pressure reading at the center. The flow in cubic feet per minute and in pounds per hour could then be calculated.

Thermocouple grids were used to measure the temperature of the air entering and leaving the furnace. Each grid was made up of 12 thermocouples located in the centers of equal areas. In the case of the grid measuring the leaving air temperature, a mass-temperature traverse was made over the 12 points and compared to the value obtained by multiplying the measured total mass flow by the average temperature. The difference was less than 1 percent, which was considered to be sufficiently accurate.

The air temperatures were taken on a recording potentiometer. The oil burned was weighed on balance scales with an accuracy of 0.05 lb. The timing of the

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* Assistant Professor of Mechanical Engineering, Queen's University.

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burner cycles was done by electric clock, and the burner was switched on and off by the operator.

The furnace selected for the initial study was an oil-fired conversion forced air unit of a *standard* design. The burner was a conventional high pressure gun-type. The furnace design referred to as *standard* consisted of an all welded steel drum with an attached crescent radiator. The radiator contained baffles which deflected the hot gases so as to make maximum use of the surface area as heat exchange surface. The furnace unit also contained the normal air circulating fan and filters.

The furnace selected for the second study was a single purpose oil-fired unit. This furnace was designed specifically for oil firing and was sold as a complete unit including the high pressure burner. The heat exchanger in this unit was also of a

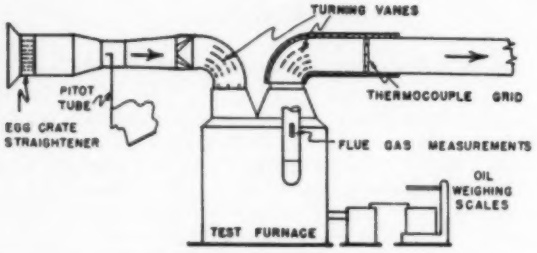

FIG. 1—SCHEMATIC DIAGRAM OF TESTING EQUIPMENT

conventional design, *i.e.* an all welded steel drum with an attached radiator encircling the drum. The unit also contained fan and filters.

The conversion furnace had 23.7 sq ft of heating surface in the main drum, and the radiator contained 22.6 sq ft of heating surface, making a total of 46.3 sq ft. The main furnace drum was 24 in. in diameter. The grate area was originally 2.4 sq ft, and the output rating at the register was given as 111,000 Btu per hr.

The oil unit had a total heating surface area of 34.7 sq ft, the central drum having 8.1 sq ft and the radiator having 26.6 sq ft. The bonnet capacity was given as 112,000 Btu per hr. The diameter of the main drum was 14 in.

To simplify the test procedure, the fan which circulated the air to be heated was run continually. The weight flow of air was held constant during the test period. The temperature of the air entering the furnace was also held at a constant value. The temperature of the air leaving the furnace therefore varied.

The heat output of the furnace was calculated for each test, using the weight flow of air through the furnace and the average temperature rise of the air during the test period. The output was expressed in Btu per hour. The heat input was calculated from the weight of oil burned during the test and the heat content of the oil. The heat input was also expressed in Btu per hour. The bonnet efficiency which was calculated for each test was the ratio of heat output to the heat input.

The controllable variables were set and held constant during all tests as follows:

1. Air flow through the conversion furnace = 1000 cfm of standard air (density = 0.075 lb per cu ft).
2. Air flow through the oil furnace = 825 cfm of standard air.
3. Smoke reading in flue gas = No. 1 set after 20 min of continuous operation.
4. Smoke pipe draft = 0.04 in. of water.
5. Conversion burner oil nozzle was a 1.5 gph with an 80 deg hollow spray cone.
6. Oil furnace burner nozzle was a 1.0 gph with a 90 deg solid spray cone.

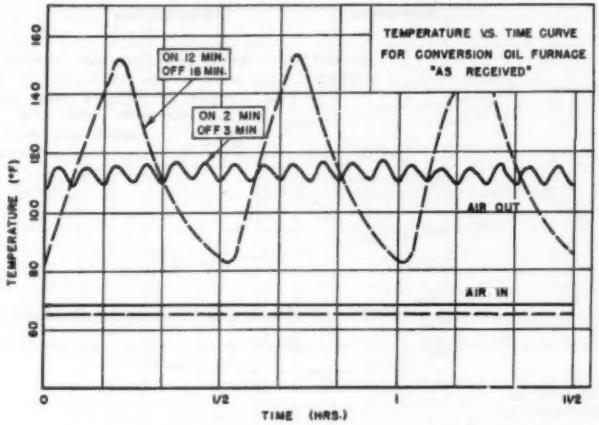

FIG. 2—TEMPERATURE-TIME CURVES AT 40 PERCENT ON-TIME CYCLES WITH CONVERSION FURNACE

Instrumentation was checked by making a heat balance during a steady-state test. The heat input was balanced against the heat output plus the flue gas loss, plus the casing loss. A check within 1 percent was obtained on this heat balance.

The conversion furnace was tested under various cycling conditions, the furnace being in the *as received* condition. The length of a cycle was varied from 5 min to 30 min in different tests, and the percent *on-time* per cycle was varied from 25 to 100. The air temperature curves for 2 tests at 40 percent *on-time* are shown in Fig. 2. The cycle length in the one case was 30 min, and in the other case, 5 min. The plot of bonnet efficiency against bonnet capacity is shown in Fig. 3, curve 1. It can be seen from this curve that the efficiency drops off as the percent of operating time decreases until a minimum efficiency is reached about 80,000 Btu per hr; then the efficiency begins to increase as the percent of operating time decreases further.

In order to investigate the reasons for the variation in efficiency in curve 1 certain conditions were changed as follows:

1. *Insulation*: It was noted that a high percentage of the casing loss from the conversion furnace occurred from the front panel which included the firing door. It was therefore decided to reduce the casing loss by insulating this front panel.

Curve 4 of Fig. 3 shows the results in efficiency of applying this insulation. It should be noted that the efficiency under continuous operation was not changed by the addition of this front panel insulation.

2. *Regenerative*: The tests referred to as regenerative in Fig. 3 were run to study the effect of heat storage capacity within the furnace. It was apparent that the heat storage capacity would affect the temperature of the leaving air during cycling operation. Fig. 4 shows the effect on air temperature of this increase in heat

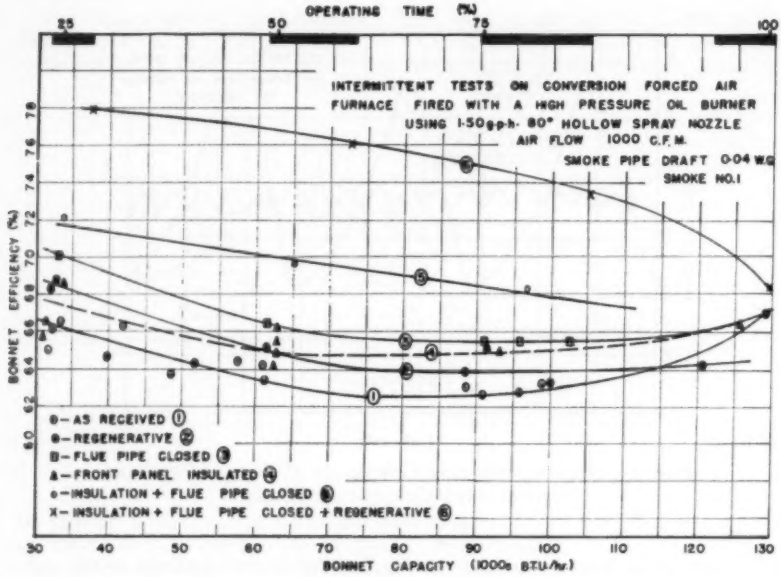

FIG. 3—RELATION BETWEEN BONNET EFFICIENCY AND BONNET CAPACITY WITH CONVERSION FURNACE

storage capacity. Curve 2 in Fig. 3 shows how the efficiency was affected. The heat storage capacity of the furnace was increased by installing firebrick on a frame over the fire inside the main furnace drum. The conversion furnace was run with 12 standard firebrick arranged near the top of the drum. The oil furnace was run with 4 standard firebrick arranged over the fire. Fewer bricks were installed in the oil unit due to the smaller space available.

3. *Flue Pipe Closed*: There was a loss of heat and hence a reduction in efficiency due to a flow of air from the smoke pipe during the off-period of the burner. In order to find the magnitude of this loss, tests were run in which the flue pipe was tightly closed during the burner off-periods. Curve 3, Fig. 3, shows the results of this test.

Curves 5 and 6, Fig. 3, show the effect on efficiency of 2 combinations of the foregoing conditions.

Fig. 5 is a plot of test results on the oil unit under various conditions of test. The cycle length was 20 min for all tests on the oil unit.

DISCUSSION

The decrease in efficiency associated with cycling operation could be due to the following conditions: (a) inefficient combustion during ignition and shut-off (b) heat loss taking place during the off-period of the cycle. Since the efficiency was

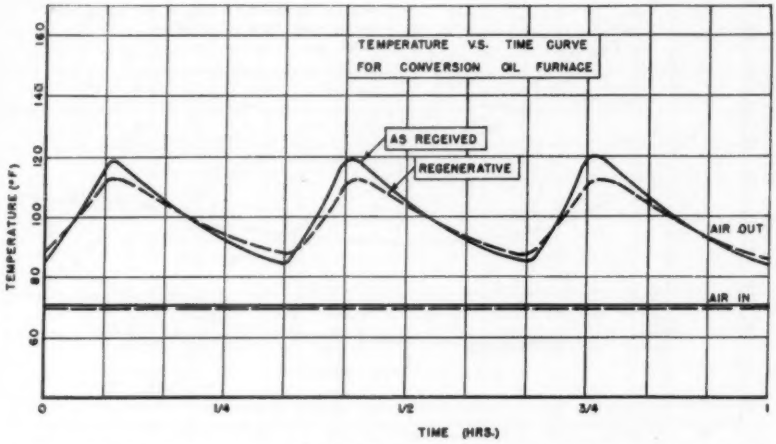

FIG. 4—EFFECT OF INCREASE OF HEAT STORAGE CAPACITY ON AIR TEMPERATURES OUT OF CONVERSION FURNACE

not affected by cycle length within the limits of these tests, the inefficient combustion during ignition would seem to be ruled out as a factor contributing to the drop in bonnet efficiency.

The heat loss taking place during the burner off-period could be broken down into casing loss and flue pipe loss. In order to show the effect of each of these losses, Curves 3 and 4, Fig. 3, were plotted. The insulation applied to the front panel did not eliminate the casing loss but only reduced it. The insulation and the closing of the flue pipe both reduced the drop in efficiency from the value found for continuous operation.

A test was then run combining the insulation and the closing of the flue pipe. The results are shown in Fig. 3, Curve 5. From the curve it can be concluded that heat losses from the furnace during the burner off-periods account for the reduction in efficiency below that of continuous operation.

Curve 5, however, indicates an efficiency increase as the operating time decreases from the point of continuous operation. This efficiency increase could only be caused by a storage of heat during the burner on-period and the release of this

heat during the off-period. In order to prove this statement, the heat storage capacity of the furnace was increased as described previously. Curve 6 shows the result of this increase in heat storage capacity.

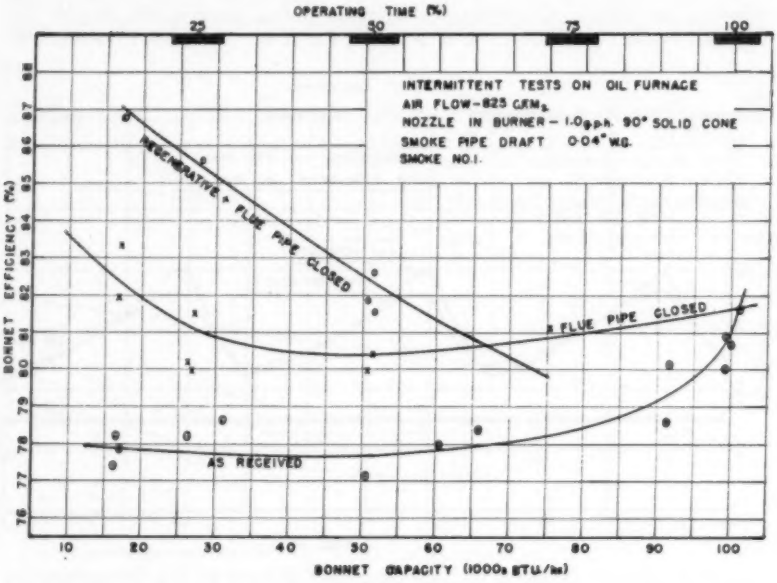

FIG. 5—RELATION BETWEEN BONNET EFFICIENCY AND BONNET CAPACITY WITH OIL FURNACE

Results of the tests on the oil unit are shown in Fig. 5. The oil unit selected for this test is representative of high efficiency furnaces, whereas the conversion furnace represents a low efficiency group.

CONCLUSIONS

1. During intermittent burner operation, the bonnet efficiency was not affected by cycle length at any given percentage *on-time* of the burner. The range of cycle length was from 5 min to 30 min.

2. The efficiency of a warm air furnace under intermittent firing conditions was found to be different from that under steady-state conditions. The difference was due to 3 factors, *viz*: (a) loss of heat from the casing of the furnace during the burner off-period, (b) loss of heat out the flue pipe during the burner off-period, (c) heat storage within the furnace during the burner on-period and the release of this heat during the off-period. Factors (a) and (b) reduce the efficiency, while factor (c) increases the efficiency. The combination of these 3 factors and the predominance of 1 over the others

will determine whether the efficiency will increase or decrease under intermittent firing as compared to continuous firing operation.

APPENDIX

ANALYSIS OF FURNACE OPERATION

Since performance characteristics have now been investigated for intermittent burner operation, an analysis of the conditions of actual operation would be useful.

Weather data from a Department of Transport weather station were compiled for 4 heating seasons, at Kingston, Canada. These data are plotted in Fig. A-1. The

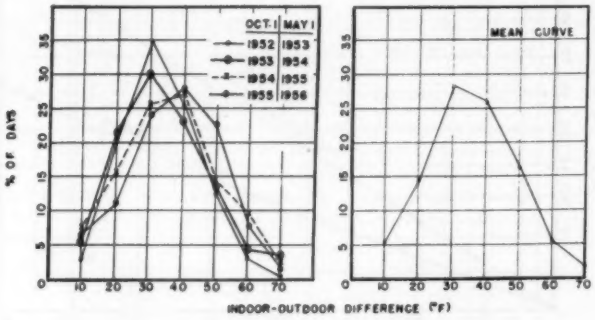

FIG. A-1—WEATHER DATA FOR KINGSTON, ONT., CANADA, DETERMINED FROM DEPARTMENT OF TRANSPORT WEATHER STATION

base axis, indoor-outdoor temperature difference, was based on an indoor temperature of 70 F. The vertical axis was the number of days in the period considered, expressed as a percent, that each indoor-outdoor difference occurred.

Fig. A-2 is a plot of running time of a burner against the indoor-outdoor temperature difference. The theoretical curve shown assumes 100 percent on-time, or continuous operation, at −10 F, which is the design temperature in use at Kingston, Canada. The actual operation line was obtained from a University of Illinois Bulletin[1] describing the study of an oil furnace in actual use in a residence. The winter design temperature at Urbana, Ill., is also −10 F.

The curves of Figs. A-1 and A-2 were combined to yield a plot given in Fig. A-3. This figure shows the number of days in the heating season as a percent, plotted against the percent operating time per cycle of the burner. It can be shown, using the actual curve from Fig. A-3, that approximately 70 percent of the time during the heating season the burner operates with an on-time per cycle between 15 and 35 percent.

This analysis gives information about the operating time of a burner when used in an actual residence. It also becomes evident that furnace performance should be con-

[1] Investigation of Oil-Fired, Forced Air Furnace Systems in the Research Residence, by A. P. Kratz and S. Konzo (*University of Illinois Bulletin, November 7*, 1939).

sidered under conditions of intermittent operation and in particular in the region of 25 to 40 percent on-time per cycle.

DISCUSSION

A. G. REED, Toronto, Ont., Canada, (WRITTEN): The results of Professor Colborne's study have revealed some thought provoking facts concerning the function of heating equipment. Intermittent operation is shown to be less efficient than continuous and this has been shown to be the result of flue pipe and casing loss during the *off* period.

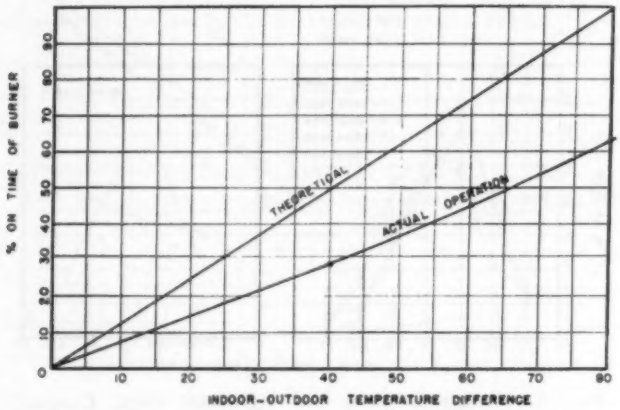

FIG. A-2—RELATION BETWEEN OPERATING TIME OF BURNER AND INDOOR-OUTDOOR TEMPERATURE DIFFERENCE. THEORETICAL CURVE ASSUMES 100 PERCENT ON TIME AT OUTDOOR TEMPERATURE OF −10 F WHILE ACTUAL CURVE WAS FROM TESTS CONDUCTED IN A RESIDENCE

Also, if a furnace can be made to store heat during the *on* period by heating up more, then the efficiency of intermittent operation can be increased.

Since the efficiency can be improved by closing off the smoke pipe during the *off* periods, it would seem that some form of automatic device for performing this function would be desirable. However, since a chimney requires a supply of heat to create draft, there may be a limit to the amount of closure. It would be interesting to know whether or not Professor Colborne used a natural-draft chimney and, if so, the draft reading over the fire just prior to the start of the burner after a shutdown period, and after the flue pipe had been opened again.

The results seem to indicate the method of rating furnaces should be changed in order to determine true efficiency of the units when installed. However, since the procedure required might greatly increase the cost of making rating tests, we wonder if research data could be developed to obtain a correction factor that would be used to determine the true bonnet efficiency in service.

J. D. BARNES, Toronto, Ont., Canada, (WRITTEN): The summary on the first page of the paper tells the purpose of the investigation. It states that the performance under

intermittent firing is different from that under continuous firing, and discusses the factors causing the changes in performance.

This paper is straightforward and leaves few points for discussion. There are, however, some questions on the technique used that do merit some discussion, and they will be outlined as headings.

Value of a Test Run Under Steady State Conditions as Compared to a Test Run Under Intermittent Firing Conditions for the Purpose of Predicting How a Furnace will Operate in an Actual Installation: The performance that must be predicted is basically to determine whether or not a furnace will deliver the heat output as rated, and continue to do

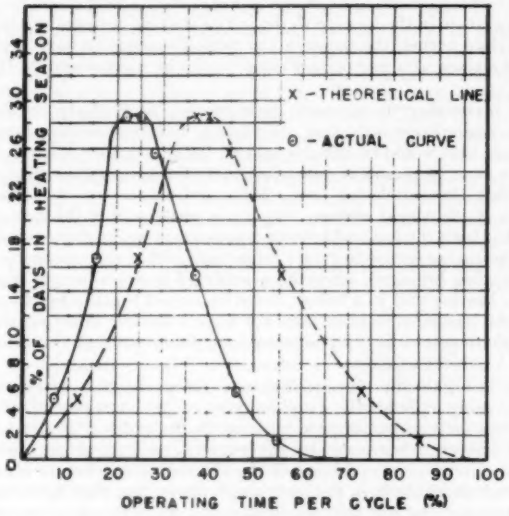
% OF DAYS IN HEATING SEASON
0 2 4 6 8 10 14 18 22 26 30 34
X - THEORETICAL LINE
O - ACTUAL CURVE
10 20 30 40 50 60 70 80 90 100
OPERATING TIME PER CYCLE (%)

FIG. A-3—RELATION BETWEEN PERCENT DAYS IN HEATING SEASON AND PERCENT OPERATING TIME PER CYCLE BASED ON CURVES OF FIGS. A-1 AND A-2

so as long as a demand for heat remains. The method of determining the required output of the furnace required to heat a home or a place of business, is to determine the heat loss from the building with a specified inside and outside temperature.

Suppose that the heating contractor has determined that 100,000 Btu per hr is the heat that is required to be supplied to a particular building to maintain this desired temperature. How then should a furnace be tested to ensure that it will deliver 100,000 Btu to this house?

Professor Colborne's paper states that a test under intermittent firing conditions will be more valuable than a test run under steady state conditions.

We are not in complete agreement with this statement and feel this point is open for a great deal of discussion.

Test Conditions: A point that is worthy of discussion is the manual method of cycling the burner with the circulating fan running continuously.

The objection to this particular method is that it does not simulate actual operating conditions of a furnace. It completely disregards any temperature extremes that may be encountered throughout the furnace, in that the normal, temperature-operated controls for both the burner and the circulating fan are eliminated.

Let us also look at the controllable variables, as they are called. The air flow is held constant which it is possible to do as is the oil input.

The smoke reading in the flue gas is equal to No. 1 after 20 minutes of continuous operation. This reading would vary from the initial start under intermittent operation. The same applies to the smoke-pipe draft.

Results: Now look at some of the results that were obtained through these tests.

Since the air-circulating fan is continuously operating, heat is continually being scavenged from the furnace, resulting as shown in Fig. 2 with a long on-off cycle in a reduction of the temperature in the furnace to the initial starting temperature between cycles. In the short cycling period, the temperature (which appears to have been taken after the furnace had arrived at a warmed-up state) varies by approximately 10 deg between cycles.

It would be interesting to compare these results, particularly on the long on-off cycle, with an intermittent-operating circulating fan temperature-controlled. One immediate change that would be noted would be the elimination of the bottom or lower half of the wave form to an extent dependent upon the temperature setting of the thermostat controlling the circulating fan. This condition is somewhat similar to that indicated in Fig. 4 when heat storage capacity is added within the furnace in which it will be noted that both the top and bottom temperature peaks are reduced. The use of insulation on the casing serves to retain more heat within the conversion-type furnace between firing cycles, but again when the question of practicability is considered in the application of a heating unit to a house, it can be argued to some length whether or not heat loss into the basement does or does not have a heating effect on the house itself. In any event, the bonnet efficiency increase over the *as received* condition amounts to approximately 4 percent as shown.

The effect of closing the flue pipe, adding heat storage capacity within the furnace and insulating the casing individually produce very slight increases in efficiency of the bonnet, but collectively a considerable increase, in the order of 10 percent at the start is noted, which difference decreases with the bonnet capacity and operating time. Again it is felt that the continuous operation of the circulating fan has an effect on these efficiencies as noted, particularly in the early stages during the short operating time cycles.

While the closing of the flue pipe produces noticeable increase in the efficiency under intermittent operation, it is very dependent on the operating time. The closing of the flue pipe or reducing the flow of air up the flue pipe during off-cycles, is not a desirable nor safe practice. Very little mention or comparison is made between the results obtained from the 2 types of heating units, although the difference is considerable.

The conversion furnace obviously releases a great deal of heat to the surrounding air which does reduce the bonnet efficiency whereas the oil-fired unit, with its insulated enclosure, shows a much higher bonnet efficiency. In some applications the former with its high heat loss to the surrounding air will be more suitable than the totally enclosed unit where heat is needed in the furnace room area.

A question arises on reading this paper, as to whether or not the author considered any other means of increasing efficiency of the furnaces besides those that are mentioned, and if so, why the 3 means mentioned were chosen. Some methods that come to mind as being practical and effective, concern combustion characteristics and the design of the combustion chamber. The choice of material and the actual dimensions of the combustion chamber do have a considerable effect upon efficiency and also on the heat storage capacity within the furnace.

The addition of bricks to increase heat storage capacity and the use of an automatic damper are not desirable nor recommended methods for increasing efficiency of furnaces. We do not feel that tests run under the intermittent conditions as described, predict how

a furnace will run in an actual installation more closely than tests run under conditions such as are specified in CSA Specification No. B140.4 and U. L. Inc. Standard 727.

R. L. WEEKS, Linden, N. J.: During the last 2 years we have engaged in studies complementary to the work that Professor Colborne has reported. Our studies have shown results which are in qualitative agreement with those he has found. We used 2 new definitions to evaluate our work; these are *system efficiency*, and what we call *relative size*. System efficiency is the efficiency of the heating system including the down-time period when, as Professor Colborne says, heat may be lost up the flue pipe. We found that this correlated extremely well with relative size.

We define relative size in these terms: A relative size of 1.0 is when the heating plant input exactly matches the heat loss of the house for the coldest day of the heating season. This is not the theoretical minimum; the minimum would be found in a fully modulated system where the firing rate matched the heat loss at all times.

We found in over 50 houses in the field that going from a relative size of 1.0 to 2.0 resulted in a decrease in system efficiency from 100 percent (by definition) for a relative size of 1 to 75 percent for a relative size of 2.0. The correlation is linear, so one can imagine the efficiency loss for houses with relative sizes of 3 and 4.

We found that a great number of homes in the Northern New Jersey area were greatly oversized, some having relative sizes approaching 4.

A solution could be had in two ways: (1) Heat losses up the flue pipe, in a house with an oversized heating plant, can be reduced by means of an automatic stack damper. A foolproof device would be needed, and I am not sure it can be made foolproof in an inexpensive way. (2) A positive sizeable saving can be had in a large number of cases by reducing the size of the firing equipment. We have recently run tests in which we deliberately cut relatively high sized homes by a factor of 2, with a resulting 20 per cent average saving in efficiency.

J. I. TRIMBLE, Toledo, Ohio: It has occurred to me that it may be of value to know about similar work we had done for us a number of years ago (1945).

We endeavored to determine the magnitude of the effect of control of *off-period* air on the overall performance of a boiler using a gas-fired conversion burner.

The reason for doing the work was that we manufactured burners which incorporated a control such that it would shut off the secondary air on the *off* cycle. This eliminated any hazard problem associated with closing off the chimney.

Our work disclosed that a warm air furnace and its cycling fan introduced more variables than we could cope with, so we made the application to a hot water boiler. We assumed that radiated heat from the boiler was going into the house and, therefore, was usable. Consequently, we confined our efforts to finding out as far as possible what was going up the stack and definitely lost.

We set up a measuring method whereby the actual volume and temperature of the air going up the stack was continuously measured for the full period of test. Various cycle times were used and, in addition, the percentage of on and off time was varied for the different cycles selected.

The result of our work was that the difference in the amount of heat that we could measure as going up the stack with the air shut off versus not shut off averaged about 1 percent; and in spite of the fact that it was the best test we could devise, the results frequently fell within the range of experimental error.

As far as we were concerned, closing off the air through the boiler did not have any practical significance in field operation.

CHARLES BARROWS, Levittown, Pa.: This is in the order of a question regarding stack loss. Since you are operating under constant draft conditions, would not the losses be greater than in the field under normal shutdown conditions?

AUTHOR'S CLOSURE: In answer to Mr. Reed's question on how we produced the draft, we made use of an induced draft fan and a rather extensive piping system throughout

the lab. The smoke pipe draft was set at 0.04 inches of water during the burner *on* period. In our particular arrangement this resulted in a smoke pipe draft of 0.02 to 0.03 inches of water during the *off* period, depending on oil rates and burner setting. These values were obtained from the *as received* tests, that is with no smoke pipe closure or added mass.

The over-fire draft was not measured since some of our work has shown that this reading has little significance.

The other point mentioned by Mr. Reed that the method of rating furnaces should be changed or that a factor should be applied to correct efficiency does not seem to me to be necessary. As far as a furnace rating is concerned I think the continuous-firing test and hence the maximum output, still is the figure most useful in the selection of a furnace for a particular job. I doubt if a factor could be found to apply to efficiency in order to yield an efficiency in service. I think each furnace will exhibit different efficiency characteristics under intermittent operation. The main points shown in this paper are, how the efficiency is affected, and what factors affect this efficiency.

The paper states that a test under intermittent firing conditions will be more valuable than a test under steady state conditions for the purpose of predicting how the furnace will operate in an actual installation. At no place in the paper is it suggested that furnace ratings should be determined any differently than by the continuous firing test. Mr. Barnes has apparently misinterpreted the purpose of this work judging from his first point.

In connection with the second topic of Mr. Barnes, the author realizes that continuous fan operation is not completely realistic. This type of operation was deemed necessary to simplify the testing. The main question is whether the results will be appreciably different with this type of fan operation. In the modern warm air system the fan is adjusted to so-called continuous air circulation, which just means that the temperature settings are low on the fan control switch. The air from the fan will therefore carry away most of the heat from the furnace. Any small heat build-up or temperature rise will cause the fan to restart even before the burner has come on. The author feels that the difference in the heat output of the furnace with the fan operating by a properly set control and with continuous fan operation, will be negligible.

As for the manual method of cycling the burner, I cannot see what criticism there could be. For the purpose of testing there must be some predetermined variables. The cycle length is one of the variables which we controlled in order to find out certain unknowns.

The author is not concerned with whether the closing of the smoke pipe is desirable or safe practice in the words of Mr. Barnes. It was not suggested that furnaces should all be equipped with shut-off dampers even though I do believe that it could be done quite safely. This was merely a method used in the testing, to prove that the heat lost up the smoke pipe during burner off period was one factor in reducing efficiency. We were also able to show to what extent this loss lowered the bonnet efficiency.

Mr. Barnes wondered if any other methods were considered as a means of increasing the furnace efficiency. Again I would point out that our purpose was not to look for methods to increase efficiency. We were analysing intermittent operation as compared to continuous operation. If increasing efficiency was our object then a look at the combustion process would be very worthwhile as Mr. Barnes suggested.

Mr. Barrows raised the question of stack loss being high due to constant draft. As I mentioned, the draft in the smoke pipe dropped to about 0.02 to 0.03 inches of water when the burner was shut off. Unfortunately this draft would be dependent on the flue system. A natural draft chimney would be desirable, however, it would introduce uncontrolled variables which we like to eliminate as much as possible. I cannot say how the *off period* smoke pipe draft would compare in our lab set-up and in a field installation with a natural draft chimney.

1614

No. 1614

MEASUREMENT OF INFILTRATION IN TWO RESIDENCES

Part I: Technique and Measured Infiltration

By D. R. BAHNFLETH*, T. D. MOSELEY** AND W. S. HARRIS†, URBANA, ILL.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS in cooperation with the University of Illinois, Urbana, Ill.

DURING the past 40 years many data on infiltration through different types of windows, doors and wall constructions have been obtained in laboratory investigations. A comprehensive bibliography can be found in the ASHAE GUIDE[1]. While laboratory studies have provided invaluable data for design infiltration heat loss estimates, their validity is frequently questioned because of assumptions which must be made to apply laboratory data to a structure having crackage on several exposures. Furthermore, in an actual structure the combined effects of wind and temperature difference cause infiltration, whereas recognized design procedures consider only the effect of wind.

Results of an investigation of infiltration in dwellings were published[2] recently, wherein the infiltration rates were measured by the tracer gas technique[3] using helium as the tracer gas. The apparent success of this method of measuring infiltration prompted procurement by the ASHAE of a helium gas analyzer.

During the winter of 1954–55 an investigation of infiltration in residences using the helium tracer gas technique was begun at the University of Illinois in cooperation with the Society. Infiltration rates were measured in the I=B=R Research Home and in Warm-Air Heating Research Residence No. 2. The results presented here were obtained in the 2 houses during the period from the winter of 1954–55 through the summer of 1956.

The principal objective was to determine the infiltration rates of the 2 houses and to correlate infiltration rates with wind velocity and direction and indoor-

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[1] Exponent numerals refer to References.
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outdoor temperature and pressure differences. An additional objective was to compare measured infiltration rates in the 2 houses with those calculated by the methods presented in THE GUIDE. The techniques used and the measured infiltration data are included in this paper. Part II will consider the effect of the several variables on the infiltration rates in the 2 houses and will include a comparison of the results of the investigations in the 2 houses.

DESCRIPTION OF RESIDENCES AND INSTRUMENTATION

I=B=R Research Home: The Research Home, shown in Figs. 1 and 2 and described[4] in detail in University of Illinois Engineering Experiment Station *Bulle-*

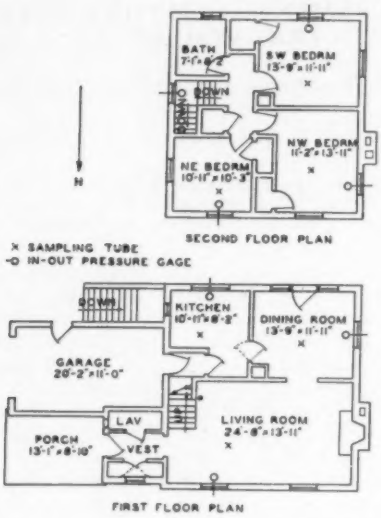

FIG. 1—FLOOR PLANS SHOWING HELIUM SAMPLING TUBE AND INDOOR-OUTDOOR PRESSURE GAGE LOCATIONS IN THE I=B=R RESEARCH HOME

tin 349, was a 2-story building with full basement typical of small well-built American homes. The total volume of the 2 stories was 9393 cu ft, and the basement volume was 5084 cu ft. Construction was brick veneer on wood frame, and all outside walls and the second-story ceiling were insulated with mineral wool batts 3⅝ in. thick. A vapor barrier was located between the insulation and the plaster base. The attic was ventilated by louvers in the gable ends. All windows were weatherstripped and of the double-hung type except for the wood casement windows in the kitchen. Storm doors were used on the 2 exterior doors during the winter. The windows and doors were all well fitted. Table 1 gives the crack

widths and lengths throughout the house. The fireplace chimney in the living room was sealed during the study. The Research Home was located on a typical city site, surrounded by trees and houses (see Fig. 2). The largest open area was a small yard to the south. With the exception of one house, located only 12 ft from the west wall, there was no other building closer than 75 ft. The surrounding houses were predominantly 2-story structures.

The Research Home was heated during the winter by a one-pipe hot water heating system. Radiation throughout most of the house was Type R cast-iron baseboard located along the exposed wall in each room. Small tube, 22-in. high radiators were used in the vestibule, lavatory, stair landing, and bath.

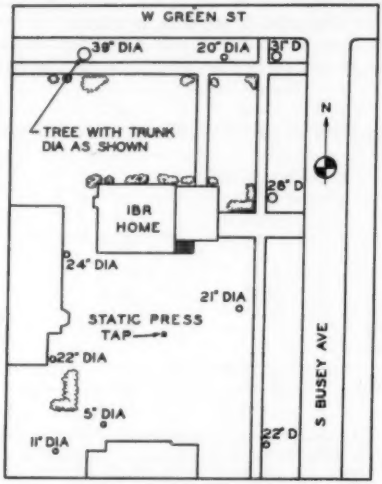

FIG. 2—SITE PLAN FOR THE I=B=R RESEARCH HOME

The chimney to which the gas-fired boiler was connected had a height of 35 ft 2 in. above the basement floor and consisted of a 6½-in. x 11-in. fireclay flue lining enclosed by one course of red brick. The smoke pipe connection was 7 ft 1 in. above basement floor level, and the draft hood was at an elevation of 5 ft 8 in. The chimney extended up through the central part of the house and was completely surrounded by heated space from its base to the level of the second-story ceiling. The plaster walls were separated from the masonry by an air space averaging about 5 in. in width. At the second story and basement ceiling levels this space was packed with mineral wool to decrease the circulation of air between the space around the chimney and the attic or basement. Another chimney of identical measurements and similar construction, but located on an exterior wall, was used for a gas water heater.

There were 2 fan-coil units, one on each story, used for summer cooling. These units were supplied with chilled water from an air-cooled chiller located outside the

house. The units were centrally located with a supply register 4 in. from the ceiling of each room. The return grille on the first story was located in the living room next to the stairs. The second floor return was in the hall. These units were used to distribute helium through the house in both summer and winter tests.

Instrumentation for determining infiltration consisted of a helium gas analyzer, a valve manifold panel and sampling tubes. A schematic diagram of the helium analyzer and the manifold panel is shown in Fig. 3. The principle of operation of the analyzer was the same as the one used by J. B. Dick[3]. The sampling tubes, which were ¼-in. OD copper tubing, were located as shown in Fig. 1, with inlets

TABLE 1—EQUIVALENT[a] CRACKAGE OF FIRST AND SECOND STORIES OF THE I=B=R RESEARCH HOME

ROOM	TOTAL LENGTH OF CRACK, FT			
	S	W	N	E
First Floor				
Living Room			37.2	
Hall			14.8	45.0
Lavatory				8.9
Kitchen	27.0			9.0
Dining Room	73.6	18.6		
Second Floor				
Bedrooms				
SW	17.8	17.8		
NE			17.8	15.9
NW		15.9	17.8	
Hall				15.9
Bath	15.9			
Total	134.3	52.3	87.6	94.7

[a] Average width of crackage for weatherstripped windows was assumed to be 0.04 in. and for weatherstripped doors, a value of 0.09 in. was assumed. Crack lengths are given in terms of an 0.04 in. crack.

60 in. above the floor in the first and second story rooms. The inlet of the attic tube was 6 in. above the joist, and that in the basement was 15 in. below the floor joists. The sampling tube in the second story hall could be moved to various locations.

Instrumentation for a study of the neutral zone location consisted of 7 differential pressure gages and one recorder, all readable to within ±0.002 in. of water. These gages were connected to outdoor total pressure taps located as shown in Fig. 1. One side of each gage was open to the room. The outdoor total pressure taps consisted of a ⅜-in OD aluminum tube which extended through a window mullion. The end of the tube was flush with the face of a small octagonal aluminum plate mounted against the mullion and parallel to the glass. The indoor-outdoor recorder measured the static pressure difference at the first-story floor level. The outdoor static pressure tap for the recorder was located 32 ft south of the house and 6 in. above the ground. The outdoor static pressure was taken from the center of the inner surface of two 12 in. square plates mounted horizontally ¼ in. apart.

Warm-Air Heating Research Residence No. 2: Research Residence No. 2 was a one-story frame structure with full basement. A detailed description has been presented[5] in University of Illinois Engineering Experiment Station *Bulletin 401*. Total volume of the first story was 8084 cu ft, including closets. Basement volume was 8890 cu ft. Walls were insulated with 3⅝-in. mineral wool blanket insulation with vapor barrier attached. The ceiling was fully insulated, and the attic space was vented by louvers in the gable ends and 1-in. slots in the eave overhang on the south and north exposures. All windows were of the horizontal sliding type, with weatherstripping, except the fixed picture window in the living room. The Residence was equipped with storm windows and storm doors during heating studies, but summer studies were made with the storm windows and doors removed. Removal of the storm windows would have no effect on the infiltration rates through the windows since they were fastened directly to the window sash. Since the doors were weatherstripped and well fitted, the change in the infiltration rate through the doors would also be small. Crackage in each exposure expressed in terms of the crack width of weatherstripped windows is listed in Table 2. The Residence was located on a comparatively open site, remote from buildings and trees. The nearest structures were a one-story weather station about 40 ft to the north, a one-story bungalow about 80 ft northeast and a row of one-story flat-roofed bungalows about 120 ft to the south.

TABLE 2—EQUIVALENT[a] CRACKAGE OF FIRST STORY OF RESEARCH RESIDENCE NO. 2

ROOM	TOTAL LENGTH OF CRACK, FT			
	S	W	N	E
Living Room	100.0			
S. Bedroom	25.0	21.5		
Bath		14.5		
N. Bedroom		25.0	22.5	
Hall			45.0	
Kitchen-Din.			41.5	17.0
Total[b]	125.0	61.0	109.0	17.0

[a] Average width of crackage for weatherstripped windows was assumed to be 0.04 in. For fixed window a value of 0.02 in. and for weatherstripped doors a value of 0.09 in. was assumed.

[b] The sum of the maximum crackages of each room was 251 ft.

During winter studies the Residence was heated by a gas-fired forced warm-air furnace. The furnace chimney was constructed of prefabricated cement-asbestos sectional units 8 in. in diameter. The total chimney height was 26 ft 11 in. above the basement floor; the 5-in. flue connection was at an elevation of 6 ft 9½ in.; and the opening to the furnace draft hood was at a height of 4 ft 3 in. above basement floor. A separate 4-in. diameter chimney of cement-asbestos material was provided for the gas-fired water heater.

During the early part of the summer a water-cooled condensing unit was used to cool the Residence. This unit was later replaced by a self-contained air-cooled condensing unit. Outdoor air was supplied to the air-cooled condenser through a 14- x 20-in. duct, and the heated air was returned to the outdoors through a duct of the same size. A basement window was replaced by a plywood panel with

openings for the condenser air ducts. The cracks between the ducts and the plywood panel and between the window frame and the plywood panel were sealed with caulking compound.

Instrumentation for these studies consisted of a helium gas analyzer, a valve manifold panel and sampling tubes (see Fig. 3). The sampling tubes, which were ¼-in. OD copper tubing, were located as shown on Fig. 4. The sampling tubes extended 60 in. above the floor in the first-story rooms, and that in the basement was 4 in. below the floor joists. The sampling tube in the return-air duct was about 1 ft below the return-air grille.

To obtain indoor-outdoor pressure differentials during the investigation, a static pressure tube[6] was located 6 ft 2 in. above the roof peak and a static pressure tap, made of ⅛-in. iron pipe, was located at the floor level in the living room. In-

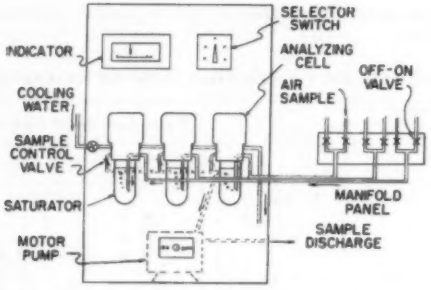

FIG. 3—SCHEMATIC DIAGRAM OF HELIUM ANALYZER AND MANIFOLD PANEL

stantaneous values of pressure differential were measured by an inclined draft gage and a continuous record was obtained by a draft recorder. The tube connecting the outdoor static pressure tap to the measuring instruments was exposed to outdoor temperature throughout its length to the point of entry into the basement at the first-story floor level. Hence, the indicated pressure differential was nearly equal to the indoor-outdoor pressure differential at the first-story floor level.

TEST PROCEDURE AND SPECIAL TESTS

Helium Release: Introduction of helium into the first and second story rooms of the Research Home was accomplished by releasing helium into the return-air plenum of each fan-coil unit through a ⅜-in. OD copper tube with 4 rows of 3/32-in. diameter holes spaced on 1½-in. centers. During the release of helium, a pressure of 10 psig was maintained in the release tube. The helium was then distributed directly into the rooms by the unit fans. The second-story rooms received the helium first. In order to give an initial concentration of about 1.3 percent, 125 cu ft of helium was released, which required about 8 min.

In Research Residence No. 2 helium was released into the return-air plenum of the warm-air heating system and drawn through the filter into the furnace casing and forced through the air distribution system into the individual rooms. The

volume released was adjusted to give an initial volumetric concentration of helium in the first-story rooms of about 1.3 percent. The time required for admitting this amount of helium (approximately 110 cu ft) was about 5 min.

Air-Helium Samples: Air-helium samples were drawn from each station through the sampling tubes, through the manifold panel, and thence to one of the 3 saturators by a pump. Samples were saturated by being bubbled through distilled water in the saturator. Thermal conductivity of the saturated air-helium sample was compared with that of a saturated air sample in the analyzer cell and was then dis-

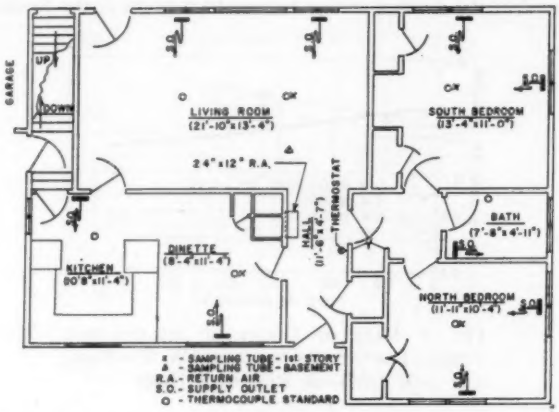

FIG. 4—FLOOR PLAN OF WARM AIR HEATING RESEARCH RESIDENCE NO. 2

charged through a ¼-in. OD copper tube into the kitchen of the I=B=R Research Home. The small quantity of helium discharged had a negligible effect on the *apparent* kitchen air change rate. In Research Residence No. 2 the air-helium samples were discharged into the return-air duct downstream from the sampling station.

Test Conditions and Data Taken: (*a*) *I=B=R Research Home*: All windows and exterior doors were closed during each test except the kitchen door leading to the basement and the outside basement door, which were opened several times during tests. The house was occupied by 2 to 5 persons, and all cabinets, drawers, and closets were opened throughout each test. During several special tests the second story was sealed off from the first.

Concentration of helium at each of the sampling stations was measured at 12 min intervals. Concurrent with the readings of helium concentration, indoor and outdoor temperature and humidity, and indoor-outdoor pressure differences at 8 locations were recorded. Wind speed and direction were obtained from the University Weather Station, located in a residential area similar to that of the Research Home. Wind velocity and direction were measured 15 ft above the roof of a 2-story building. Duration of the tests was from 1 to 1½ hours.

(*b*) *Warm-Air Heating Research Residence No. 2*: During each study all outside doors and windows were closed. The blower was operated continuously to assure

uniform mixing of the air and helium in the rooms, and operation of the gas burner or the refrigeration compressor was controlled by the thermostat on the first story. During most of the studies the living quarters were not occupied. All cabinets, drawers, and closets were closed throughout each test except during 2 special studies made with closets open.

Concentration of helium at each of the sampling stations was measured at 6 min intervals. Wind velocity and direction, indoor and outdoor temperature and humidity, velocity pressure and draft in the smoke pipe, and the indoor-outdoor pressure difference were recorded during each test. Wind velocity and direction were measured at the weather station located 40 ft north of the Residence. The

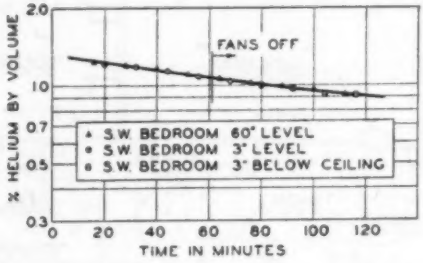

FIG. 5—RATE OF DECAY OF HELIUM FOR THREE ELEVATIONS IN THE SW BEDROOM OF I=B=R RESEARCH HOME

anemometer and direction indicator were located approximately 23 ft above ground level. Duration of the tests varied from 1½ to 2 hr.

Distribution Tests: The rate of infiltration determined by the tracer gas technique is valid only when the tracer gas is uniformly distributed in the room. British studies[2] have shown that erroneous *apparent* air change rates could be obtained due to stratification of the tracer gas. Since helium is considerably lighter than air, the validity of taking samples at one point in each room was questioned and several distribution tests were run in each of the houses.

Helium concentration was measured at various levels and in corners of 2 rooms in the I=B=R Research Home. Fig. 5 shows results of a test in the southwest bedroom where the concentration was measured at 3 levels—3 in., 60 in., and 7 ft 6 in. There was practically no difference in concentration at any of these levels, and even after the fans were turned off there was excellent agreement. Since this test was conducted in August, the question of room temperature gradients may arise. However, during summer cooling the gradients were as severe as during winter heating.

Similar results were obtained in Research Residence No. 2. During each study, however, the fan was operated continuously, and no attempt was made to determine the amount of stratification with the fan off.

In the studies in Research Residence No. 2 it was observed that the helium decay curves, when plotted on semi-logarithmic paper, deviated slightly from the predicted straightline relationship. This deviation was attributed to leakage of helium into the closed closets and exterior walls and to absorption by furnishings.

To determine the effect of closets on the decay of helium concentration, several studies were made with the closets opened but with the drawers and cupboards closed. Comparison of the data obtained on similar days with and without the closets closed showed that similar deviations occurred. From this it was concluded that the effect of the closets was not the dominant cause of the observed deviation.

House Stack Effect Test: When temperature difference is the only force causing infiltration there is a flow of air into a building near the ground level and out at a higher elevation; this flow reverses when there is a higher temperature outside the building than inside. Flow of air between the first and second stories in the Research Home was studied by placing a seal between the 2 stories, injecting helium into one story and allowing it sufficient time to mix well, then removing the seal and noting the changes in helium concentration throughout the house.

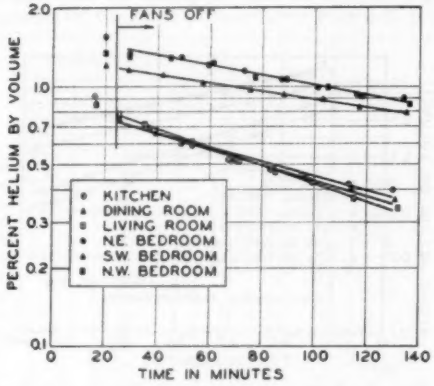

FIG. 6—RATE OF DECAY OF HELIUM WITH FANS NOT OPERATING IN THE I=B=R RESEARCH HOME

When helium was released on the second story very little ever reached the first story after removing the seal, but when helium was added initially to the first story a rapid increase was noted on the second story after removing the seal. The results show that most but not all air change between first and second stories was upward during the heating season.

RESULTS

I=B=R Research Home: If it is assumed that the decay of helium concentration per unit time for the whole Research Home is equal to the helium leaving the Home with the exfiltrating air in the same unit time, then

$$-V\,dc = nV\,c\,dt$$

where

V = house volume.
c = instantaneous concentration of helium.

n = number of air changes per unit time.
t = time

the equation when solved gives

$$n = ln\ (c_1/c_2)/(t_2 - t_1)$$

$$\text{or equivalently } c_2 = c_1\, e^{-n(t_2-t_1)}$$

Thus, on semi-logarithmic paper the decay curves will be straight lines, the slopes of which are directly proportional to the infiltration rate. Typical decay curves for the first and second-story rooms of the Research Home are shown in Fig. 6. Except for the mixing period, 20–30 min, the points fell very close to straight lines.

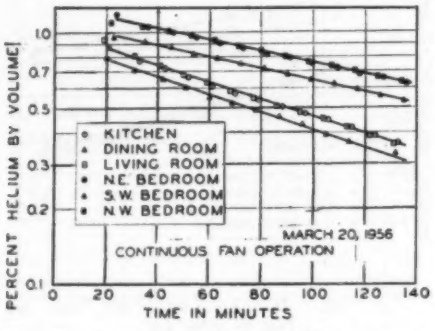

FIG. 7—RATE OF DECAY OF HELIUM WITH CONTINUOUS FAN OPERATION IN THE I=B=R RESEARCH HOME

There was some deviation which may be partly attributed to helium leaking into walls, furniture, and similar places when the concentration was high and then returning to the room air when the concentration had decreased.

The steep living room decay curve was the result of several factors. The vestibule, with a large amount of crackage area, opened directly into the living room. This condition was accentuated when the heating plant was in operation because of the removal of air from the basement through the chimney. Basement air was partly replaced by first story air leaking through cracks around the kitchen door leading to the basement. This caused a greater air change on the first story than when the chimney was cool. The same reasoning can be applied to the dining room in view of the large crackage area there also.

Varying slopes of decay curves in a single story occurred only when the house air was relatively still. Fig. 7 shows that when the fans were operated continuously the slopes of decay curves on any one story were approximately the same. One sampling tube on each story would have given accurate results with the fans in operation. However, provided there is no helium stratification, still air conditions within the house will give a more accurate account of infiltration in individual rooms than when the air is stirred by fans. The slopes of the decay curves for the 2 stories shown in Figs. 6 and 7 should not be interpreted to mean that the infiltra-

tion in the first story is greater than that of the second story. The movement of helium-laden air from the first to the second story would cause the *apparent* air change of the second floor to be less than the actual air change rate.

The infiltration rate for the Research Home was increased very slightly when the fans were operating. This was due to leakage of air from the second story fan unit plenum into the attic. For this reason all test results given here are with fans not operating.

Table 3 gives a summary of the weather conditions during each test. The indoor-outdoor static pressure difference at the first-story floor level and the location of

TABLE 3—SUMMARY OF TEST CONDITIONS—I=B=R RESEARCH HOME

TEST NO.	DATE	TEMPERATURE		IN-OUT TEMP. DIFF., F	AVG. WIND VELOCITY, MPH	MAX. WIND GUST, MPH	WIND DIRECTION	IN-OUT PRESS. DIFF., IN. H_2O	NEUTRAL ZONE FROM FIRST FLOOR, IN.
		IN-DOOR F	OUT-DOOR F						
1	2–10–56	75	28	47	4	13	S	−0.018	164
2	2–28–56	75	24	41	8.5	15	NW	−0.018	155
3	3– 3–56	73	51	22	7	23	N	−0.015	167
4	3– 8–56	73	25	48	13	27	W	−0.021	169
5	3–15–56	73	39	34	9	17	NE	−0.013[a]	160[a]
6	3–20–56	73	42	31	6	15	S	−0.014[a]	139[a]
7	4–10–56	73	46	27	9.2	20	E	−0.012[a]	145[a]
8	4–16–56	73	43	30	9.7	34	NW	−0.017	158
9	4–19–56	74	61	13	8.3	21	N	−0.009	122
10	5– 3–56	74	70	4	5.5	15	S	−0.005	115
11	6–27–56	76	83	−7	9.5	22	W	..	..
12	7– 5–56	76	71	5	6.2	12	NW	..	..
13	8– 6–56	77	82	−5	7.5	19	N	..	..
14	8–14–56	75	84	−9	6.7	17	N	..	..

[a] Data not shown in Figures.

the neutral zone above the first-story floor level are also given; these values have been omitted for the last 4 tests because of inaccuracy of the pressure measuring instruments. The pressure difference and neutral zone for test No. 10 are questionable for the same reason.

Infiltration rates of the house and individual rooms for all tests are given in Table 4. The rates for the house and rooms were omitted for several tests because they were not representative of actual infiltration rates.

Warm-Air Heating Research Residence No. 2: The procedure followed in the analysis of the data obtained in Research Residence No. 2 was the same as that just outlined. The number of air changes at each of the sampling stations on the first story was obtained from the slopes of the helium decay curves plotted on semi-logarithmic paper. A summary of the weather conditions during each test is given in Table 5, and the air change rates calculated from the decay of helium concentration during each test are tabulated in Table 6. The average air change rates listed in Table 6 were computed as the summation of the individual air change rates weighted with respect to room volume. Since the weighted average rate and that measured in the return-air duct differed by only a small amount, the data observed in the return-air duct have been used to represent the air change rate of the entire first story.

TABLE 4—SUMMARY OF MEASURED INFILTRATION RATES—I=B=R RESEARCH HOME

TEST NO.	NO. OF AIR CHANGES CALCULATED FROM TESTS						
	LIV. RM.	DIN. RM.	KITCH.	NE BED.	NW BED.	SW BED.	HOUSE[d]
1	0.43	0.33	0.29	0.27	0.26	0.24	0.32
2[a]	Special Test						0.38
3[b]	Special Test						0.35
4[b]	Special Test						0.43
5[a]	Special Test						..[e]
6[c]	0.48	0.47	0.44	0.31	0.31	0.31	0.40[e]
7[c]	0.08	0.07	0.09	0.34	0.32	0.32	0.26[e]
8	0.35	0.25	0.33	0.38	0.42	0.38	0.35
9	0.32	0.24	0.29	0.25	0.26	0.27	0.28
10	0.12	0.17	0.15	0.15	0.14	0.33	0.17
11	0.10	0.10	0.11	0.19	0.24	0.22	0.16
12	0.13	0.22	0.18	0.25	0.28	0.21	0.20
13	0.12	0.17	0.28	0.23	0.12	0.16	0.16
14	0.17	0.20	0.43	0.14	0.08	0.16	0.17

[a] *Special Test*—House stack effect test with helium release on first story.
[b] *Special Test*—House stack effect test with helium release on second story.
[c] *Special Test*—Continuous fan operation during test.
[d] The house air change rate was computed from a weighted average of the room rates.
[e] Data not shown in Figures.

TABLE 5—SUMMARY OF TEST CONDITIONS—RESEARCH RESIDENCE NO. 2

TEST	DATE	TEMPERATURE		IN-OUT TEMP. DIFF., F	AVG. WIND VELOCITY, MPH	MAX. GUST, MPH	WIND DIRECTION	IN-OUT PRESS., DIFF., IN. H_2O
		INDOOR F	OUTDOOR F					
1	12-22-54	73.4	32.0	41.4	9	15	SSW	−0.001
2	12-28-54	73.5	32.0	41.5	10.5	21	N	0.003
3	1-11-55	73.4	31.4	42.0	5	10	SSE	−0.012
4	1-14-55	73.8	32.0	41.8	12	29	S	−0.008
5	1-28-55	73.9	3.9	70.0	7	11	WNW	−0.010
6	2-10-55	74.6	20.6	54.0	14	28	NNW	0.028
7	3- 2-55	74.7	55.2	19.5	8	17	SSE	0.006
8	3- 7-55	73.3	24.7	48.6	8	20	WSW	0.006
9	3- 8-55	74.4	60.0	14.4	15	27	WSW	0.027
10	3- 9-55	74.9	64.8	10.1	13	22	WSW	0.024
11	3-14-55	73.8	64.2	9.6	10	21	SSE	0.010
12	3-17-55	73.0	32.4	40.6	5	8	SE	−0.008
13	5- 6-55	77.8	80.8	− 3.0	8	17	S	0.010
14	5-19-55	80.0	72.0	8.0	3	5	W	−0.001
15	6-29-55	75.4	86.1	−10.7	7	15	SSW	0.011
16	7- 8-55	75.1	93.7	−17.6	8	20	SW	0.016
17[a]	7-20-55	75.0	87.0	−12.0	5	13	E	0.008
18[a]	7-21-55	77.7	89.9	−12.2	4	8	SE	0.007
19[a]	7-25-55	78.2	87.5	− 9.3	4	13	S	0.005
20[a]	7-26-55	76.2	93.5	−17.3	5	10	SW	0.009
21[a]	8- 5-55	76.2	86.9	−10.7	5	9	SW	0.007
22[a]	8- 5-55	76.0	88.0	−12.0	5	12	SW	0.008
23[a]	8- 8-55	75.0	76.2	− 1.2	5	11	SE	0.005

[a] Tests 17 through 23 were made with self-contained air-cooled condensing unit installed in Residence. Leakage of condenser air to recirculation air passage increased infiltration and tests have not been used in analysis.

Tests 1 through 14 in Table 5 were taken during the heating season and with the exception of Test 13 the indoor temperature was greater than that outdoors. Tests 15 through 23 were taken during a summer cooling investigation when the indoor temperature was less than that outdoors.

Of the 9 tests conducted during the summer, only the first 2 have been included in the results. The last 7 tests, which were made with the self-contained air-cooled condensing unit installed, have been omitted because of increased infiltration resulting from leakage of outdoor air used for condensing into the recirculated room air. The condition of the air-cooled unit during each of the 7 tests is given in the last column of Table 6.

TABLE 6—SUMMARY OF MEASURED INFILTRATION RATES—RESEARCH RESIDENCE NO. 2

TEST NO.	NO. OF AIR CHANGES CALCULATED FROM TESTS						NOTES
	N BRM	DIN RM	S BRM	LIV RM	RETURN	AVE.	
1	0.59	0.56	0.56	0.55	0.57	0.56	
2	0.57	0.54	0.54	0.56	0.54	0.55	
3	0.54	0.50	0.49	0.52	0.53	0.52	
4	0.63	0.59	0.60	0.60	0.63	0.60	
5	0.93	0.76	0.77	0.78	0.80	0.80	
6	0.62	0.54	0.59	0.56	0.60	0.57	
7	0.45	—	—	0.32	0.33	—	
8	0.50	—	0.50	0.50	0.49	—	
9	0.48	0.38	0.43	0.38	0.40	0.41	
10	0.41	0.38	0.40	0.36	0.38	0.37	
11	0.32	0.34	0.36	0.31	0.36	0.33	
12	0.41	0.39	0.44	0.41	0.44	0.41	
13	0.27	—	0.27	—	0.28	—	
14	0.28	0.26	0.27	0.25	0.26	0.26	
15	0.36	0.32	0.37	0.34	0.33	0.34	1
16	0.38	0.36	0.38	0.37	0.36	0.37	2
17[a]	0.59	0.54	0.52	0.54	0.51	0.54	3
18[a]	0.63	0.64	0.64	0.64	0.64	0.64	4
19[a]	0.37	0.36	0.37	0.36	0.37	0.36	5
20[a]	0.45	0.43	0.43	0.42	0.42	0.43	3
21[a]	0.63	0.56	0.56	0.56	0.56	0.57	6
22[a]	0.33	0.34	0.37	0.34	0.37	0.35	3
23[a]	0.34	0.37	0.35	0.37	0.36	0.36	7

[a] These studies were not included in comparisons with the winter data.

Notes: 1—Water-cooled, unit off; 2—Water-cooled, unit on; 3—Air-cooled, unit on; 4—Air-cooled, unit off, closets open; 5—Air-cooled, duct sealed, closets open; 6—Air-cooled, unit off; 7—Air-cooled, duct sealed

SUMMARY

1. The Research Home, which was located on a typical city site surrounded by large trees and houses, had a considerably smaller infiltration rate at a given wind velocity measured above tree top level than that of Research Residence No. 2, which was located on a relatively open site, free of trees, and typical of new housing developments.

2. The helium tracer gas technique could be utilized for field measurements of infiltration in a variety of houses, thus providing much needed information on the applica-

tion of laboratory data to actual construction. In houses equipped with warm-air heating systems, a single helium analyzing cell located in the total return-air stream would provide satisfactory indication of the house infiltration rate. In houses equipped with hot water or steam heating systems, a single cell located on each floor would give satisfactory results provided some means were used to keep the room air well mixed.

ACKNOWLEDGMENTS

Acknowledgment is made to the *Institute of Boiler and Radiator Manufacturers* and to the *National Warm Air Heating and Air Conditioning Association* for permitting their respective research staffs to participate in these studies and for allowing the studies to be conducted in their experimental houses.

The authors wish to thank H. Hagan, formerly visiting scholar from Norway, and C. F. Chen, formerly research assistant in Mechanical Engineering, for the collection and analysis of much of the data obtained in Warm-Air Heating Research Residence No. 2. The authors also wish to thank F. J. Poskocil, student assistant, for the collection of much of the data obtained in the I=B=R Research Home and for preparing many of the drawings in this paper.

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DISCUSSION

Discussions of this paper are combined with Part II. To find them, turn to page 466.

1615

No. 1615

MEASUREMENT OF INFILTRATION IN TWO RESIDENCES

Part II: Comparison of Variables Affecting Infiltration

By D. R. BAHNFLETH*, T. D. MOSELEY** AND W. S. HARRIS†, URBANA, ILL.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS in cooperation with the University of Illinois, Urbana, Ill.

INFILTRATION measured in the I=B=R Research Home and Warm Air Heating Research Residence No. 2 by the helium tracer gas method and the weather conditions under which the measurements were made were presented[1] in Part I of this paper. The Research Home was a 2-story brick veneer house with full basement, located on a typical city site surrounded by houses and trees. Research Residence No. 2 was a single-story frame structure with full basement, and located on an open site typical of many new housing developments. Differences in location and construction of the 2 houses caused several noticeable differences in the results of the 2 investigations. The objective of this part of the paper is to compare the effects of the variables related to infiltration in the individual houses and to compare the results for the 2 houses.

Since the measured data were obtained over a range of combinations of wind velocity and direction, and indoor-outdoor temperature difference, and since only a limited number of tests were conducted in each house, it was deemed advisable to correct the measured data to specific conditions of wind and temperature difference. The corrections to the original data were determined from the slopes of plots of the original data assuming in all cases that a straight-line relationship existed between the dependent and independent variables. For example, the average increase in infiltration with each mph increase of wind velocity was ob-

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[1] Exponent numerals refer to References.
Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Murray Bay, Que., Canada, June 1957.

tained from a plot of air change rate as a function of wind velocity. This incremental change was used to correct the measured air change rates to conditions of 0, 7.5 and 15.0 mph winds. The corrected air change rates were plotted as a function of indoor-outdoor temperature difference and the new curves of air change *vs.* temperature difference were used to obtain correction factors for temperature difference. The corrections were repeated twice to obtain the second approximation. Although an inspection of the data for the 2 houses shows that wind direction had some effect on the results, it was not possible to draw conclusions from the data and no corrections were made for wind direction. Unless otherwise noted,

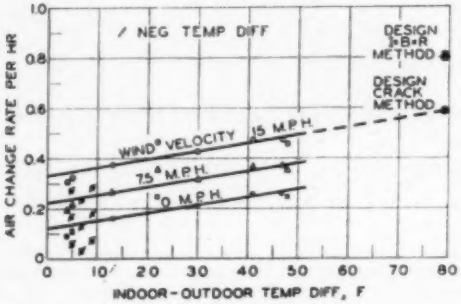

FIG. 1—EFFECT OF TEMPERATURE DIFFERENCE ON FIRST- AND SECOND-STORY AIR CHANGE RATE IN THE I=B=R RESEARCH HOME

the data have been corrected by the foregoing procedure to separate the effects of wind velocity and temperature difference.

RESULTS IN I=B=R RESEARCH HOME

The house air change rate is shown in Fig. 1 as a function of indoor-outdoor temperature difference corrected by 2 approximations for 0, 7.5 and 15 mph wind. The scattering of points is due partly to the wind data, which was taken from the University Weather Station and measured above the roof level where no trees or buildings obstructed it.

Wind velocity affected the infiltration as shown in Fig. 2. Infiltration rate increased with an increase in wind velocity. Wind direction may have an effect on infiltration provided there is unequal crackage on various sides of a building. In the Research Home it was not possible to correlate wind direction and infiltration. It can be seen from Figs. 1 and 2 that a change of 4 deg F in the temperature difference was equivalent to about a 1 mph change in wind velocity. Wind velocity had a special effect on the Research Home by increasing air flow up the chimney; this will be discussed later.

All points used in constructing the curves of Figs. 1 and 2 were obtained when the indoor temperature exceeded the outdoor temperature. Points with slashes

represent a negative temperature difference and were not considered in drawing the curves. They indicate that the summer infiltration rate for a given temperature difference and wind velocity was lower than that for winter. This discrepancy can be explained by the effect of the chimney on increasing the house air change rate during the winter, as discussed later, and the additional shielding during the summer of leaves on the trees surrounding the Research Home.

When infiltration rate is estimated it is common practice to assume that no air passes through properly sealed wall construction. Infiltration rates for the Research Home, as calculated by the *crack* and *air change* methods, are shown in Figs.

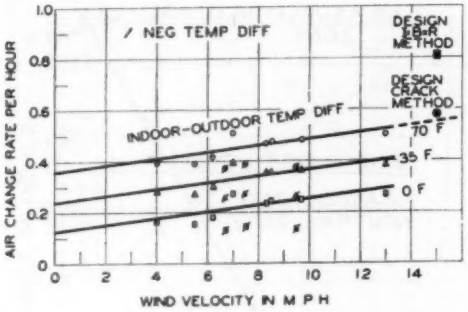

FIG. 2—EFFECT OF WIND VELOCITY ON FIRST- AND SECOND-STORY AIR CHANGE RATE IN THE I=B=R RESEARCH HOME

1 and 2. The crack width for windows and doors was estimated from material in THE GUIDE[2] 1956. The crack was assumed to be 0.04 in. for the weatherstripped windows and 0.09 in. for weatherstripped doors. For these calculations it was assumed that there was no air change between the house and attic and the house and basement. It is interesting to note that the infiltration rate estimated by the *crack* method and the *air change* method fall into good agreement with the test data. However, this is coincidental since the design value was based on wind velocity and did not consider temperature difference.

When investigating infiltration rates it is helpful to consider the neutral zone or plane. The neutral zone is that elevation in a building where the pressure indoors is equal to that outdoors when the building is exposed only to an indoor-outdoor temperature difference. Air flows into the building below the neutral zone and out above it when the temperature indoors is greater than that outdoors. The flow pattern is reversed when the outdoor temperature exceeds the indoor temperature. In the Research Home the neutral zone was calculated to be 100 in. above the first-story floor level. To make this calculation is was assumed that there was no air change between the attic or basement and first and second stories, and that there was no air flow through the exposed walls. In order to determine the approximate location of the neutral zone, the measured indoor-outdoor pressure difference at several locations was plotted against elevation (Fig. 3). A negative

pressure difference indicates that the indoor pressure was less than that outdoors. The letters beside each point indicate the exposure where the total pressure was measured. There may be a different neutral zone for each exposure when a wind is blowing, but Fig. 3 gives a mean neutral zone since the curve was drawn taking

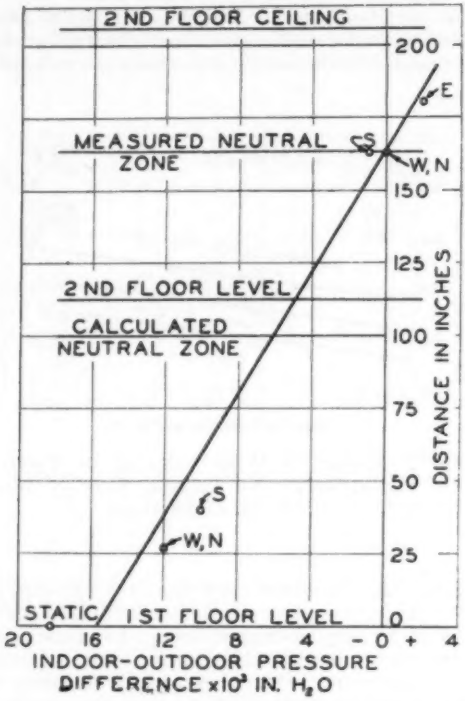

FIG. 3—LOCATION OF THE NEUTRAL ZONE AS DETERMINED BY THE INDOOR-OUTDOOR PRESSURE DIFFERENCE AT VARIOUS ELEVATIONS IN THE I=B=R RESEARCH HOME WITH SOUTH WIND AT 4 MPH, FEBRUARY 10, 1956

all points into consideration. A straight-line relation was assumed; however, this did not introduce a large error because the zero pressure difference usually occurred near the second-story pressure measuring level.

The indicated location of the neutral zone is shown in Fig. 4 as a function of the indoor-outdoor static pressure difference at the first-story floor level. The height of the neutral zone above the first-story floor increased with increased pressure difference, and the measured height of the neutral zone was always greater than

that estimated from the crackage on the 2 floors. The difference in the estimated and measured neutral zone locations might be attributed to leakage through the second-story ceiling not considered in the calculation of the neutral zone location. However, if the ceiling did cause the difference, the neutral zone location should have occurred at some fixed point above the estimated location since the resistance to flow would be constant. Since the height of the neutral zone location varied during the tests another cause of the variation was suggested. The difference was attributed to the effect of the heating plant chimney.

In cold weather the chimney caused a partial vacuum in the basement which was replaced partly by air from the first story; this prevented some of the first-story in-

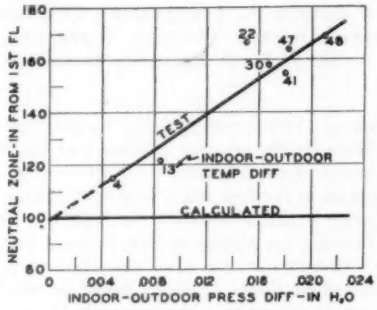

FIG. 4—EFFECT OF INDOOR-OUTDOOR PRESSURE DIFFERENCE (FIRST FLOOR LEVEL) ON LOCATION OF NEUTRAL ZONE IN I=B=R RESEARCH HOME

filtrating air from leaving the house through the second-story windows, giving an *effective* greater crack area in the second story than in the first story. The numbers by the points in Fig. 4 are the average indoor-outdoor temperature differences which give an indication of the heating plant operation. The chimney alone could cause a rise of the neutral zone above the second-story ceiling by creating a partial vacuum in the entire house.

Tests indicated air leakage from the house into the basement and into the attic. The slight attic leakage was probably around the fan-coil unit even though it was enclosed in a plastic seal. Smoke tests confirmed leakage of air through the kitchen door cracks into the basement.

Effects of indoor-outdoor temperature difference and wind velocity on indoor-outdoor static pressure difference at the first-story floor level during the heating season are shown in Figs. 5 and 6. The data have been corrected by the second approximation for 0, 35 and 70 deg F temperature differences and 0, 7.5 and 15 mph wind velocities. The curves show that a 1 deg F rise in temperature difference is equivalent to about a ¾ mph increase in wind velocity on the indoor-

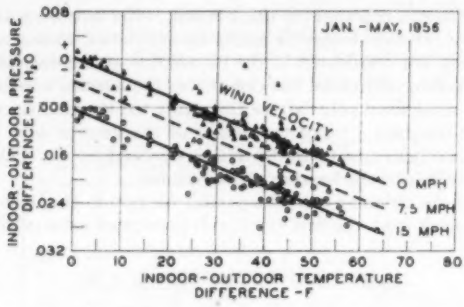

FIG. 5—EFFECT OF TEMPERATURE DIFFERENCE ON INDOOR-OUTDOOR PRESSURE DIFFERENCE IN THE I=B=R RESEARCH HOME

outdoor pressure difference. It is significant that temperature difference and wind velocity had similar effects on the pressure difference at the first-story floor level. That is, increases in temperature difference or wind velocity, caused a reduction in the indoor static pressure at the first-story floor level, making the indoor-outdoor pressure difference a larger negative amount. In buildings exposed to wind pressures, the infiltration through the windward side causes a pressure build-up inside the building which is a function of the wind velocity and the ratio of the crack areas on the windward and lee sides. Thus, it would be concluded that the wind had little direct effect on the pressures in the Research Home. However, the wind had an indirect effect on the pressures because of its effect on the air flow in the heating plant chimney. Further discussion of the chimney flow will follow later in this paper.

To determine the relationship between indoor-outdoor pressure difference and house air change rate, Fig. 7 was plotted using the static pressure difference at the first-story floor level. It can be seen that the house infiltration rate increased

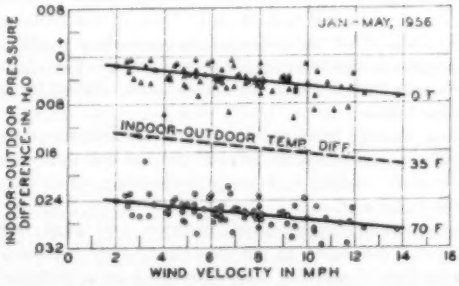

FIG. 6—EFFECT OF WIND VELOCITY ON INDOOR-OUTDOOR PRESSURE DIFFERENCE IN THE I=B=R RESEARCH HOME

rapidly for a given change in indoor-outdoor pressure difference at the first-story floor level at a small pressure difference, but slowly for a larger pressure difference. The slope of the house air change—pressure difference curve would be different if the reference pressure difference were measured at another location. If the pressure difference had been measured nearer the neutral zone, but still beneath it, the slope of the curve would have been much greater. The limiting case would occur when the pressure difference was measured at the null point—the neutral zone. For this case, the vertical axis would represent the relationship of air change and pressure difference.

The effect of the heating plant chimney on the infiltration and pressure difference has been referred to frequently. Air and combustion gases leave the Research

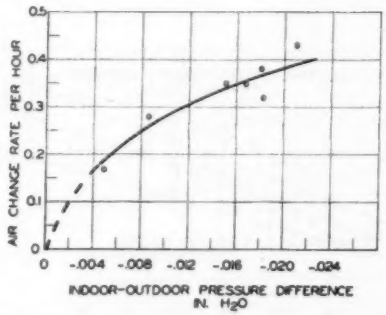

FIG. 7—EFFECT OF INDOOR-OUTDOOR PRESSURE DIFFERENCE ON FIRST AND SECOND STORY AIR CHANGE RATE IN THE I=B=R RESEARCH HOME

Home through the chimney at the rate of about 11,400 cu ft per hr[3] (0.8 air changes for the house including the basement) with an indoor-outdoor temperature difference of 35 deg F. The rate varied slightly depending upon temperature difference. It was estimated that the crackage of windows and doors in the basement and a separate flue for a gas water heater could supply about 90 to 95 percent of the air removed by the heating plant chimney. By comparing the winter and summer points on Figs. 1 and 2 it appeared that the combined effect of the chimney and bare trees increased the total infiltration by about 0.1 air changes during the winter as compared to summer conditions.

Effects of the chimney on the indoor-outdoor pressure difference at the first-story floor level are further shown by Figs. 5 and 6. It can be seen from Fig. 5 that with an increase in wind velocity there was an increase in indoor-outdoor pressure difference. This was the result of wind causing an additional air flow up the chimney, thus creating a greater suction in the basement which had a like, but lesser, effect on the first and second stories. The increase in air flow up the chimney caused by wind, in excess of that caused by temperature difference, ranged from 0 to 4000 cu ft per hr for an 8 mph wind, and from 3000 to 5000 cu ft per hr for

a 15 mph wind. Thus, for a 15 mph wind the total flow in the chimney ranged from 14,000 to 16,000 cu ft per hr.

It may, therefore, be concluded that infiltration rates determined for the Research Home were slightly high due to leakage into the attic and basement, predominantly the basement. This explains the rise of the neutral zone with an increase in indoor-outdoor pressure difference, Fig. 4. The Research Home infiltration rate changed about 0.013 air changes per hr for each 4 deg F increase in indoor-outdoor temperature difference or 1 mph increase in wind velocity.

WARM AIR HEATING RESEARCH RESIDENCE NO. 2

Effect of indoor-outdoor temperature difference on infiltration rate is shown in Fig. 8. The original data have been corrected by the same method just described

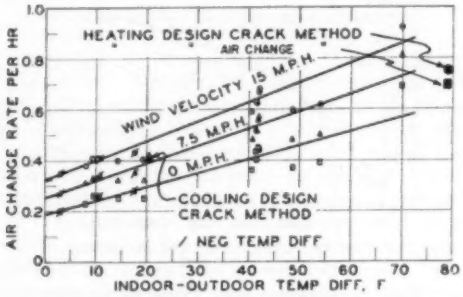

FIG. 8—EFFECT OF TEMPERATURE DIFFERENCE ON FIRST STORY AIR CHANGE RATE IN THE WARM-AIR HEATING RESEARCH RESIDENCE NO. 2

for wind velocities of 0, 7.5 and 15 mph. The data from days on which the temperature indoors was less than that outdoors have been distinguished by a slash. For each deg F change in indoor-outdoor temperature difference the infiltration rate increased approximately 0.0066 air changes or 53 cu ft per hr. The summer infiltration rates (indicated with a slash) were of the same order of magnitude as the winter rates for the same absolute value of indoor-outdoor temperature difference.

The effect of wind velocity on the infiltration rate is shown in Fig. 9 where the number of air changes is plotted as a function of wind velocity with the data corrected as before for temperature differences of 70, 35 and 0 deg. The cup-type anemometer used to obtain wind velocity during these studies was located about 23 ft above the ground level atop the weather station 40 ft north of the Residence. The recorded wind velocity may, therefore, be greater than the velocity at the level of the windows. The wind velocity at window level would be less than that measured at the anemometer and a correction for the difference would cause an increase in the slope of the air change curves. For each 1 mph increase in wind velocity the infiltration rate increased about 0.012 air changes or 97 cu ft per hr.

Part of the scatter of the points in Fig. 9 may be attributed to the effect of wind direction. Since the major part of the crack area was in the south and north walls, the highest air change rates would be anticipated when the wind was blowing from either of these directions. It was, however, impossible to draw any conclusions regarding the effect of wind direction.

The infiltration rate of the first story was estimated by the *crack* method and by the *air change* method using data given in THE GUIDE[2] 1956. In the *crack* method estimate, infiltration was assumed to occur in each room of the Residence without regard to wind direction. The length of crack used for computing the infiltration rate in each room was taken as the maximum crackage on one exposure. The total equivalent crack length used was 251 ft, or approximately 80 percent of the total equivalent crackage of the first story. The infiltration through the walls was assumed negligible. The estimated infiltration rates for design conditions during

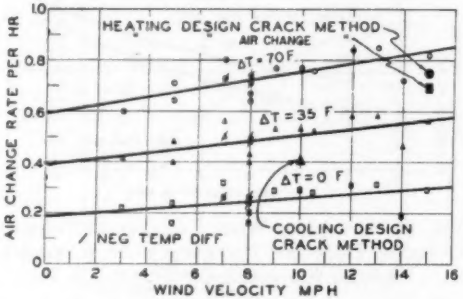

FIG. 9—EFFECT OF WIND VELOCITY ON FIRST STORY AIR CHANGE RATE IN WARM-AIR HEATING RESEARCH RESIDENCE NO. 2

heating and cooling have been plotted on Figs. 8 and 9. Design heating and cooling air change rates, which were estimated by the *crack* method and based on wind velocities of 15 and 10 mph, respectively, were 0.75 and 0.40 air changes. The design heating air change rate estimated by the air change method was 0.69 air changes. The maximum winter air change rate of 0.8, which was observed on a day when the outdoor temperature was about 0 F and the wind velocity 7 mph, was approximately 7 percent greater than that estimated by the *crack* method and 16 percent greater than that estimated by the *air change* method. The maximum measured summer air change rate of 0.36, which was observed on a day when the outdoor temperature was 95 F and the wind velocity 8 mph, was 10 percent less than the design rate. Hence, within the limits of the assumptions made in the estimate, the agreement between the design air change and that measured when design conditions were approached can be considered good. It should be noted, however, that the good agreement was obtained by over-estimating the infiltration due to wind forces and by neglecting the infiltration due to temperature difference forces. Whether similar agreement would be obtained in situations where the wind forces are predominant is problematical.

The difference between indoor and outdoor static pressure at the first-story floor level was recorded during all infiltration tests. The outdoor pressure tap was 6 ft 2 in. above the peak of the roof, and assumed to be outside the pressure region caused by the flow of air around the house. The tubing connecting the tap to the draft gage was exposed to outdoor temperature to its point of entry at the first-story floor level. Hence, the recorded pressure differentials were equivalent to the static pressure difference at the floor level. The average pressure differentials observed during each test are plotted as a function of wind velocity in Fig. 10. Since the data showed a considerable deviation from a straight-line relationship, no attempt was made to make corrections to specific conditions of wind and temperature difference. However, the available data fell on curves of temperature difference ranging from 8 to 20 F in one case and 40 to 55 F in the other. The

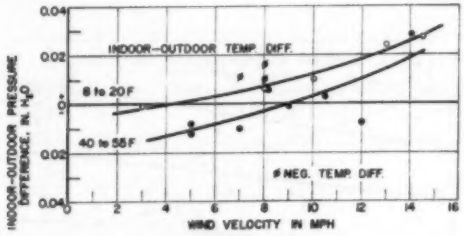

FIG. 10—EFFECT OF WIND VELOCITY ON INDOOR-OUTDOOR PRESSURE DIFFERENCE IN WARM-AIR HEATING RESEARCH RESIDENCE NO. 2

pressure build-up anticipated in structures exposed to wind forces occurred in Residence No. 2. Since the wind and temperature difference forces were acting simultaneously, one causing negative pressure differentials and the other causing positive pressure differentials, the resultant pressure difference was positive only when the effect of wind became dominant. As the temperature difference increased, the wind velocity required to cause positive pressure differentials at floor level also increased.

Results of a previous[4] investigation showed that wind direction had an appreciable effect on the pressure differentials at floor level because of different crackages in the exposures of the house. In earlier studies, the effects of temperature difference were not considered, but the trends established for each exposure were similar to those of Fig. 10. With the wind blowing from some directions, namely the east, the pressure differences were always negative during the winter because of the small amount of crackage in the east exposure.

As mentioned previously, the choice of the location of the reference pressure influences the slope of the curves of house air change as a function of pressure difference. When both wind and temperature difference forces influence the reference pressure, as was the case in Research Residence No. 2, a family of curves would be required to show the relationship of total infiltration and pressure difference at the floor level. At zero pressure difference, the slopes of the curves would change from negative to positive when moving in a positive direction along the pressure

difference axis. Since the data were limited, it was impossible to determine the relationship of infiltration and static pressure difference at the first-story floor level. Using the crack data for the first story and basement, calculation shows that the neutral zone is located approximately 30 in. above the first-story floor. For this estimation it was assumed that no air was passing through the walls or the first-story ceiling, that the resistance to air flow from the basement to the first story was negligible and that no air was leaving through the chimney.

With the crack areas around doors and windows known and the neutral zone located, it was possible to compute the relationship between indoor-outdoor temperature difference and wind velocity giving the same amount of infiltration in the

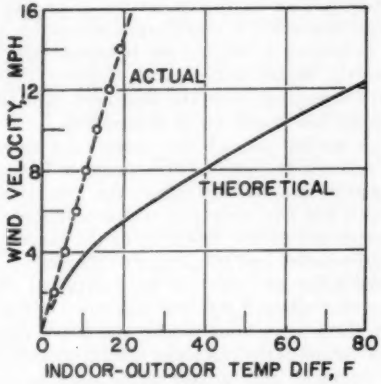

FIG. 11—THEORETICAL AND ACTUAL RELATION BETWEEN INDOOR-OUTDOOR TEMPERATURE DIFFERENCE AND WIND VELOCITY GIVING THE SAME INFILTRATION RATE

first story. In this estimate it was assumed that infiltration due to wind forces would occur through one-fourth of the total crack of the first story and that due to temperature forces would occur through the total crack of the residence located below the estimated neutral zone. The flow through the cracks was assumed to vary with the square root of the pressure difference. This relationship is shown in Fig. 11 together with the actual relationship determined from the zero wind velocity and zero temperature difference curves of Figs. 8 and 9, respectively. Whereas the theoretical relationship indicates that the wind is the dominant effect, the results of the studies indicate clearly that the effect of temperature difference was more pronounced.

This difference can be explained if it is assumed that the flow resistance of the first-story ceiling is comparatively low. A low resistance in the ceiling would raise the neutral zone, and thus increase the amount of infiltration caused by temperature difference. It would, however, have very little influence when the wind is the

only active force. It should be noted that the ceiling of Research Residence No. 2 is constructed of pressed cement-asbestos boards with taped joints. Even though no cracks were apparent, it was not possible to inspect the tightness of all joints. Recorded wind velocity and indoor-outdoor pressure difference data show that with zero wind there was as much suction at the floor level of the first story as theoretical calculations show when the neutral zone is assumed to be just at the ceiling level.

Further explanation of the difference is found in the effect of the flow of gases in the chimneys. A considerable amount of air was leaving the basement through the chimneys during winter, both combustion air for the furnace and the water heater and air entering the smoke pipes through the draft diverters. The volume varied somewhat with outdoor temperature, but was an average of 4500 cu ft per hr, or approximately half an air change for the basement. Since this air was taken directly from the basement, it would seem reasonable that it would be replaced with outdoor air leaking in around the basement windows and air leaking down from the first story. Smoke tests showed, however, that during the heating season, air was always streaming from the basement into the first-story rooms. The magnitude of this air flow could not be determined.

As the amount of air leaving through the chimney increases, the suction in the basement increases, causing a decrease in the amount of basement air entering the first story. Consequently, the chimney causes the neutral zone to shift upward in the same manner as a low flow resistance in the first-story ceiling. It is likely that both factors contributed to the difference of the theoretical and actual relationships between wind velocity and temperature difference. The extent to which each factor affected the difference could not be ascertained from available data.

Although no helium was released into the basement during the studies, within a short period of time there was a considerable amount of helium in the basement air. During the winter studies, the basement helium concentration was attributed solely to leaks in the ductwork and furnace casing since, as was mentioned before, smoke tests showed that air was always streaming from the basement to the first story. During the summer studies, a portion of the helium in the basement air could be attributed to leaks in the ductwork; however, since the concentration of helium increased to a much higher level than had been noticed during the winter studies another source of helium was suggested. Since the theoretical location of the neutral zone was estimated 30 in. above the first-story floor, it was assumed that the air would flow from the first story to the basement. Smoke tests confirmed this assumption. This mixing of basement air and first-story air by leakage from the air distribution system and by gravitational effects had some influence upon the rate of decay of helium concentration in the first-story rooms. Consequently, there may be minor errors in the air change rates calculated from the observations.

COMPARISON OF RESULTS FOR THE TWO RESIDENCES

When comparing the results from the 2 houses it should be noted that the houses are similar in size and have almost equal crackages. The volumes of the 2 houses, excluding the basements, differ by only 1309 cu ft, the Research Home having the greater volume. The distribution of the cracks on the 4 exposures is almost the same in the 2 houses.

Although the houses are similar, their infiltration rates differ considerably, especially at high wind velocities and high temperature differences. With a 15

mph wind and a 70 deg F temperature difference, the infiltration rate of Research Residence No. 2 is about 50 percent higher that that for the Research Home. This difference can be explained by considering several factors. The Research Home is well protected from wind; this decreases the wind effects on infiltration. Wall and ceiling construction of the Research Home (plaster) is tighter than that of Residence No. 2 (plywood and pressed cement-asbestos board panels fastened with screws); this decreases the effects of both wind and temperature difference.

The tight construction of the Research Home and the protection from wind is partially offset by the stack effect of the house. Since the Research Home is 2 stories high and Residence No. 2 is one story, there is a greater potential for air flow into and out of the Research Home than in Residence No. 2.

Difference in exposure to wind was shown by the measured indoor-outdoor pressure difference for the 2 houses. In the Research Home the wind increased the vacuum at the first-story floor level through its effect on the chimney draft. In Research Residence No. 2 a pressure build-up occurred at floor level when the effect of wind was greater than that of temperature difference. The effect observed in Residence No. 2 would be anticipated for a structure exposed to wind forces.

The heating plant chimney had a much greater effect on the infiltration in the I=B=R Research Home. The flow of gases in the chimney was on the average equal to 0.8 to 1.0 air changes for the entire house including the basement. Most of the air flow in the chimney infiltrated through cracks around the basement windows and the vent on a gas-fired hot water heater. However, the chimney caused the pressure difference at the first-story floor level to be greater than predicted through its effect on the location of the neutral zone. In Residence No. 2 the flow of chimney gases was equivalent to 0.27 air changes for the first story and basement, and the chimney apparently had a small effect on the infiltration. The influence of the chimney caused air to stream from the first story to the basement in the Research Home, but in Residence No. 2 the effect of the chimney was so small that the natural flow from the basement to the first floor was not interrupted.

As was mentioned previously, the validity of the tracer gas technique for measuring infiltration was questionable because of the extremely high diffusion rate of helium. It is conceivable that helium diffuses through the walls, ceilings and floors, thus indicating a higher than actual infiltration rate. The bottom curves of Figs. 1, 2, 8 and 9 suggest this fact since the curves do not pass through the origin, but intersect the air change axis, and indicate infiltration when there is no driving force (zero wind velocity and zero temperature difference). Furthermore, at zero wind velocity and zero temperature difference, the air change rate for the Research Home is about 0.12 and that for Residence No. 2 is about 0.19; this denotes that there would be more diffusion through the walls of Residence No. 2 than through the walls of the Research Home. It is apparent that helium would diffuse more readily through the walls of Residence No. 2 than through the walls of the Research Home if it is assumed that the wall construction resistance to water vapor diffusion is an index of its resistance to helium diffusion.

CONCLUSIONS

1. Good agreement between measured and calculated air change rates was obtained. The good agreement was a result of over-estimating the effect of wind forces and neglecting the effect of temperature difference forces. In the *crack* method the maximum in-

filtration in each room is used to obtain the total infiltration for a structure. Since the wind can act on only one or two exposures, the design precedure leads to an over-estimate of the total infiltration caused by wind. If the 2 houses studied had been located in other climates, the agreement between measured and calculated infiltration rates would not have been as good because the temperature difference was an important factor in both houses.

2. The Research Home, which is located on a typical city site surrounded by large trees and houses, has a considerably smaller infiltration rate at a given wind velocity measured above tree top level than that of Research Residence No. 2, which is located on a relatively open site, free of trees, and typical of new housing developments.

3. The flow of flue gases in the chimneys of the 2 houses during the winter had an appreciable effect on their infiltration rates. The effect was more pronounced in the Research Home, which is a 2-story structure.

4. Loss of helium by diffusion through walls and ceiling may have caused the *apparent* air change rates to be greater than the actual rates.

ACKNOWLEDGMENTS

Acknowledgment is made to the *Institute of Boiler and Radiator Manufacturers* and to the *National Warm Air Heating and Air Conditioning Association* for permitting their respective research staffs to participate in these studies and for allowing the studies to be conducted in their experimental houses.

The authors wish to thank H. Hagan, formerly visiting scholar from Norway, and C. F. Chen, formerly research assistant in Mechanical Engineering, for the collection and analysis of much of the data obtained in Warm-Air Heating Research Residence No. 2. The authors also wish to thank F. J. Poskocil, student assistant, for the collection of much of the data obtained in the I=B=R Research Home and for preparing many of the drawings in this paper.

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DISCUSSION

T. C. MIN, Cleveland, Ohio (WRITTEN): The authors are to be commended upon this contribution to the information on infiltration in residences. During the past year, research on entrance infiltration in multi-story buildings was undertaken at the ASHAE Research Laboratory. The results of the project are now being prepared into a report. It will be of interest to the members of the ASHAE to know that some of the findings, as pointed out by the authors, are very similar to those conducted at the Laboratory.

In Fig. 5, Part II, of the paper, at a wind velocity of 15 mph and zero indoor-outdoor temperature difference, the outdoor-indoor pressure differential is 0.008 inch of water which corresponds only to a wind velocity of less than 4 mph. Also in Fig. 6, Part II, at the zero indoor-outdoor temperature curve, the outdoor-indoor pressure differentials vary approximately only 0.004 in. of water for wind velocities from 2 to 14 mph.

FIG. A—VIEW OF THE KATHAROMETER INSTALLED

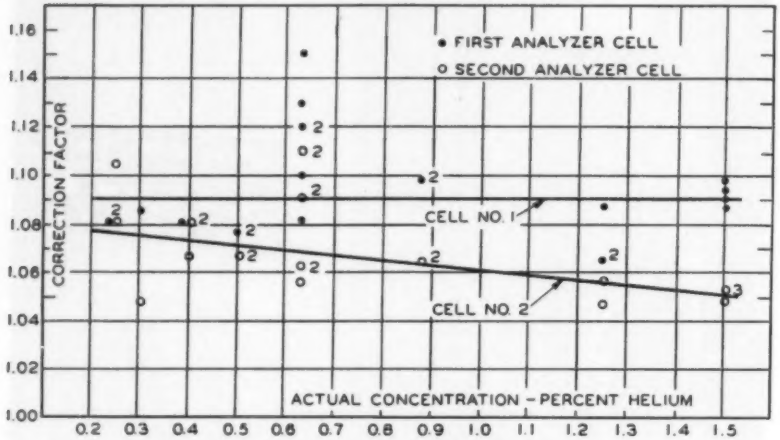
In the field tests of the ASHAE entrance infiltration study, it was found on some very windy days with wind velocities of more than 20 mph, according to the local weather station's report, that the wind velocities measured windwardly at the curb of the street near the building entrance were not more than 6 mph. The wind in the direction perpendicular to the entrance was not more than 2 mph, which is equivalent to a pressure differential of less than 0.002 inch of water. The difference in wind velocities between the weather station's reading and the test value is obvious since the former observed the wind at the top of the building in an open space such as an airport, while the latter determined the velocity under conditions with obstructions, especially in the congested downtown area.

There are also 2 minor points which the writer would like to check with the authors:

1. When the outdoor-indoor pressure differential is zero, is it true that the neutral zone could be any place instead of only one place, 100 in. above the floor as indicated in Fig. 4, Part II?

2. Is it correct that when the time increases, the concentration of helium decreases? In other words, is it in order that a negative sign may be placed in the equation $Vdc = nVc\,dt$ in Part I?

C. W. COBLENTZ,†† Washington, D. C. (WRITTEN): A commercial helium katharometer purchased by the American Society of Heating and Air-Conditioning Engineers



Numerals next to points indicate number of observations
Correction factor equals Actual Concentration/Instrument Reading

FIG. B—CALIBRATION OF THE ASHAE KATHAROMETER

was loaned to the National Bureau of Standards for an evaluation of its performance and for studies of air infiltration in the Test Bungalow at the Bureau. The instrument is a thermal conductivity meter designed to indicate concentrations of helium in air in the range of zero to 1.6 percent. Since the thermal conductivity of helium is approximately 6 times that of air, mixtures of helium and air, even in low concentrations of helium, have a measurably greater thermal conductivity than pure air. Thus by using helium as a tracer gas a katharometer of this type can be used to measure air leakage into buildings or other closed spaces. A photograph of the instrument is shown in Fig. A.

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†† It is to be noted that the discussion presented here by Mr. Coblenz is a paper entitled *Performance Characteristics of the ASHAE Katharometer*, by C. W. Coblenz, P. R. Achenbach, and E. M. Tierney, all of the staff of the National Bureau of Standards.

Description of the Instrument: This instrument is equipped with 3 separate thermal conductivity cells so that 3 samples of helium-air mixture can be drawn simultaneously for concentration determinations. These samples are drawn through the apparatus by a motor-driven vacuum pump and the source of each sample was determined by an arrangement of tubes and stopcocks shown at the right in Fig. A.

Because the thermal conductivity of air, or of an air-helium mixture is affected by its moisture content, the samples were bubbled through water and were practically saturated when reaching the analyzer. The gas bubblers can be recognized under each of the cells on the photograph, which also shows the helium indicator at the upper left of the instrument.

Each of the thermal conductivity cells contains 2 platinum resistance elements which were heated with a constant electric current and which form 2 legs of a Wheatstone bridge circuit. The resistance elements of each cell were installed in separate cavities drilled in a brass block so the wall temperatures of the 2 cavities would be equal. One

TABLE A—NUMBER OF MEASUREMENTS AND MAXIMUM, MINIMUM, AND MEAN VALUES OF CORRECTION FACTORS

ACTUAL CONCENTRATION	NO. OF OBSERVATIONS MADE		CORRECTION FACTORS DETERMINED					
			MAXIMUM		MINIMUM		MEAN	
	CELL #1	CELL #2	CELL #1	CELL #2	CELL #1	CELL #2	CELL #1	CELL #2
0.633	6	7	1.149	1.100	1.083	1.056	1.117	1.081
1.504	4	4	1.098	1.052	1.086	1.048	1.092	1.051

cavity was hermetically sealed; the other cavity had openings at each end to permit the passage of the gas sample. Thus, when a sample containing helium was drawn through one cavity, the resistance element in that cavity was cooled below the temperature of the other in the sealed reference cavity. This cooling effect lowered the resistance of the wire causing an imbalance of the Wheatstone bridge circuit which was measured by a galvanometer. Since the cooling effect is directly related to the percentage of helium present in the sample, the galvanometer can be calibrated in percentage of helium.

The selector switch, shown in the upper right of Fig. A, permitted rapid observation of the helium concentration in any of the 3 samples passing through the katharometer, after initially flushing out the sampling lines.

The rate of flow through the 3 cells can be regulated individually by the needle-valve controls shown in the photograph at the left of each bubbling chamber. The cells were equipped with water jackets to prevent over-heating of the cells and to maintain a steady temperature of the cells. These water jackets are shown connected in series in the photograph. Water flow through the jackets can be controlled by the valve at the left of the photograph. Water was not passed through the jackets during the tests of the instrument since the room air temperature was controlled by a thermostat.

Accuracy of Helium Concentration Indications: Tests were made to determine *the accuracy* of the instrument in indicating the absolute concentrations of helium in air. Pure helium was taken from a high-pressure storage cylinder and transferred into water-sealed flasks where it was held at atmospheric pressure. The displacement method was used to transfer the helium from these flasks to water-sealed sampling flasks and

helium-air mixtures were prepared in concentrations ranging from 0.3 percent to 1.6 percent helium. Several samples of each concentration were passed through the katharometer and the correction factor for each measurement was determined as the ratio of the known helium concentration to the indicator reading. Fig. B shows the correction factors plotted against the known concentrations. Fig. B shows considerable scattering of the individual observations for a given helium concentration. A correction factor curve has been drawn for each of the 2 cells used although it is recognized that the plotted observations do not define any precise line.

The effect of scattering on the correction factors determined for actual concentrations of 1.504 percent and 0.633 percent is shown in Table A which gives the number of measurements made with 2 cells of the apparatus and the maximum, minimum, and mean values for the correction factors.

TABLE B—EFFECT OF CORRECTION FACTOR OF CELL #2

HELIUM CONCENTRATION		CORRECTION FACTOR		AIRCHANGE		ERROR, %
INITIAL	FINAL	INITIAL	FINAL	APPARENT	ACTUAL	
1.5	0.522	1.050	1.071	1	0.981	1.9
1.2	0.442	1.057	1.073	1	0.985	1.5
0.9	0.331	1.063	1.076	1	0.988	1.2
0.6	0.221	1.070	1.078	1	0.991	0.9

It will be noted in Table A that the maximum deviation of the values determined for the correction factor at an actual concentration of 0.633 percent was 0.066 for the first cell and 0.044 for the second cell, whereas for an actual concentration of 1.504 percent the corresponding deviations were only 0.012 and 0.004. The deviations in indicated helium concentration for a given actual concentration are believed to be caused in part by variations in the heat transfer from the resistance elements to the brass blocks as discussed in the following section.

A constant correction factor would not cause an error in an infiltration rate determination. Thus, if the horizontal line shown for cell No. 1 in Fig. B were considered to be the true average performance of that cell, no error in infiltration determinations would result from the fact that the indicated value was 9 percent below the actual concentration. A correction factor curve of the slope shown for cell No. 2 in Fig. B would cause errors as shown in Table B for one complete airchange in a space.

Drift of Zero Adjustment and Parallax: The position of the hand on the helium indicator fluctuated around the zero point as much as 1½ scale divisions, corresponding to 0.03 percent helium, during a period of 7 hours when a flow of pure air was initiated through the cells. The drift of the indicator appeared to depend in part on whether or not a steady state of heat flow existed between the platinum resistance elements in the cells and the enclosing brass block. When air flow was initiated through the cells after having the heating elements energized without gas flow overnight, the zero position of the indicator hand changed about one scale division for each cell during the first hour of operation. During the next 6 hours the indicator position on cell No. 2 changed an additional half scale division in the same direction whereas that for cell No. 1 changed a half scale division in the opposite direction. One and a half scale divisions corresponds to an error of 2 percent in the absolute value at 1.5 percent meter reading, 5 percent at 0.6 percent meter reading, and 15 percent at 0.2 percent meter reading.

Errors in reading the curved indicator scale caused by parallax are considered to be of the order of one-fourth scale division, or 0.005 percent helium. Thus, parallax

could cause a 2½ percent error in the reading for a concentration of 0.2 percent helium.

Uniformity of Response of Three Cells: In order to determine whether or not the 3 cells were identical in their response, samples of a helium-air mixture were drawn continuously from the same station in the Test Bungalow for a period of about 110 minutes and passed through all 3 cells of the instrument while the helium concentration decreased from an initial value of about 1 percent to a final value of about 0.2 percent. Because the Test Bungalow is completely enclosed in an insulated enclosure, equal temperatures could be maintained inside and outside the building. Also, the air conditioning blowers in the space surrounding the building produced a constant air velocity outside the house.

TABLE C—UNIFORMITY OF RESPONSE OF THREE CELLS TO IDENTICAL AIR-HELIUM MIXTURES

	CELL #1	CELL #2	CELL #3
First period (0 to 72 min)			
Initial Concentration, %	0.998	0.965	0.885
Final Concentration, %	0.500	0.492	0.449
Airchange Rate	0.576	0.560	0.563
Ratio of Airchange Rate, %, to that of Cell 2	102.4	100.0	100.3
Second period (80 to 110 min)			
Initial Concentration, %	0.370	0.384	0.343
Final Concentration, %	0.182	0.194	0.171
Airchange Rate	1.416	1.364	1.380
Ratio of Airchange Rate, %, to that of Cell 2	103.8	100.0	101.3

Therefore, the infiltration rate of the building could be maintained steady over an extended period of time. Inside and outside temperatures and the air movement in the enclosure were not changed during the entire test. In order to obtain an accelerated infiltration rate for a portion of the test, the exhaust fan in the attic of the Bungalow was operated during the last 30 minutes of the test.

Fig. C shows the observed helium concentrations plotted against time on semilogarithmic paper and the best-fitting straight line drawn for each cell. After tracer-gas measurements had been taken at 5 minute intervals for a period of 70 minutes, the attic fan was turned on. The resulting change of the infiltration rate is indicated by the change in slope of the 3 nearly parallel lines. Some of the individual concentration values deviate from the straight line which is the criterion for a constant infiltration rate. Although there is a maximum difference of 14 percent in the helium concentration indication among the different cells, the difference between the computed infiltration rates was much smaller, as shown in table C.

Table C shows the initial and final concentrations given by the lines fitted to the data for each cell and for the 2 parts of the test. The airchanges which occurred during these periods were calculated as the natural logarithms of the ratios of the helium concentrations at the beginning and at the end of the period. The airchange rate during the first 72 minutes of the test computed from Cell No. 3 was 0.3 percent higher than for Cell No. 2, and the airchange rate computed from Cell No. 1 was 2.4 percent higher than for Cell No. 2. For the second period of the test the corresponding differences were 1.3 and 3.8 percent, respectively. The best-fitting lines shown in Fig. C for Cells 2 and 3 are nearly parallel thus indicating that the infiltration rate computed from these 2 cells would be nearly the same.

Effect of Gas Pressure at the Cell: The instrument was found to be practically insensitive to atmospheric pressure fluctuations or pressure differences that may arise from

long sampling lines. A test showed that a change in pressure at the inlet of the instrument from 36 in. of water positive pressure to 36 in. of water vacuum increased the indicator reading by only 0.02 percent of helium at the middle of the scale.

Effect of Sampling Flow Rate: A test was made to determine the effect of the gas-flow rate through the bubblers on the indicator reading. For this purpose an orifice flow meter was installed in the sampling line and the helium-air mixtures were drawn from water-sealed flasks containing 0.45 and 0.65 percent of helium in air, respectively. The results of this test are plotted in Fig. D, which shows the correction factor for air flow rates from 40 to 378 cubic centimeters per minute.

Fig. D shows an increasing correction factor for gas flow rates in the range from 40 to 250 cc per min and a decreasing correction factor for flow rates above 250 cc per min.

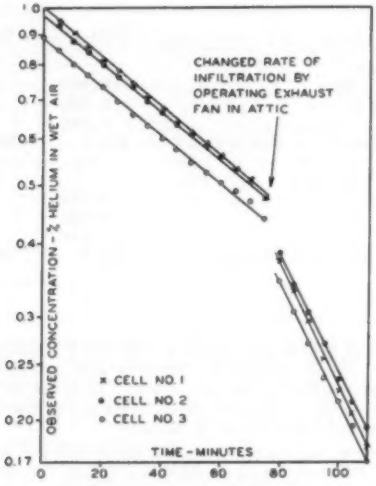

FIG. C—RESPONSE OF THREE CELLS UNDER IDENTICAL CONDITIONS

A correction factor of unity was observed for a flow rate of about 125 cc per min. There was some scattering of the observed values, but the relation between correction factor and flow rate was similar for the 2 levels of helium concentration used. The curve fitted to the plotted values shows only one percent change in correction factor in the range of flow rates from 175 to 350 cc per min.

Effect of Length of Sampling Line: The time lag of the katharometer was determined by alternately introducing a helium-air mixture, from a 6-gallon water-sealed can, and pure air from the room into 2 of the test cells and averaging the readings during 5 such cycles. The results of these tests with samples drawn through 3 ft of ¼-in. copper tubing are shown in Fig. E. Fig. F shows the result of a similar test using 30 ft of ¼-inch tubing. It will be noted that near balance was reached after 3.5 minutes with the 3 ft sampling line, whereas with a 30 ft line the indicator required about 5 minutes to arrive at near balance. Thus, the time constants were about 55 seconds and 2 minutes for the 3-foot and 30-foot sampling lines, respectively.

Conclusions: The characteristics of the specimen katharometer described indicate that the following practices should be followed during use to minimize errors in infiltration measurements.

(1) Air should be drawn through the cells for at least an hour before any measurements are taken.

(2) Helium concentrations in the range from 0.4 to 1.4 percent should preferably be used to reduce the importance of the error caused by drift and parallax.

(3) The rate of gas flow through the bubblers should preferably be kept constant, although variation in the range from 200 to 350 cc per min was shown to cause an error of one percent or less.

(4) The shortest possible sampling lines should be used and appropriate consideration given to the lag of the instrument at the beginning of a test.

(5) Numerous readings of concentration should be taken during any test to average out the errors caused by drift and parallax.

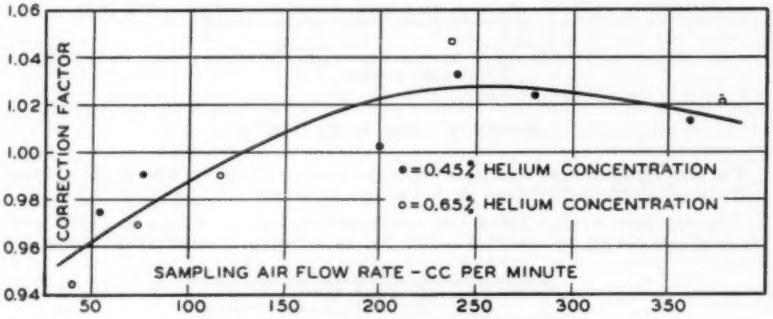

FIG. D—EFFECT OF SAMPLING AIR FLOW RATES

(6) The use of a decay range in the lower third of the indicator scale would result in lower errors from the slope of a correction factor curve such as shown for Cell No. 2. However, this advantage would probably be more than offset by the greater error resulting from drift in the lower third of the scale.

Observing the precautions summarized will minimize the errors in results caused by drift of the indicator needle, parallax, and variability of gas flow. This study of the characteristics of the ASHAE katharometer indicates that it is a satisfactory instrument for infiltration measurements.

R. A. PARSONS, Lansing, Mich., (WRITTEN): These two papers provide interesting information on a subject that needs clarifying. More good work of this type should be encouraged. My comments are presented for the purpose of obtaining more effective use of the data.

Conclusion 4 of Part II regarding loss of helium by diffusion through walls was not considered in comparing measured and estimated air change rates. Furthermore, the first sentence of Conclusion No. 1 of Part II, regarding good agreement between measured and estimated air change rates, is questionable when the data is examined carefully, and when adjusted for the 0.19 changes per hour due to diffusion as in the accompanying Table D for Warm Air Heating Research Residence No. 2.

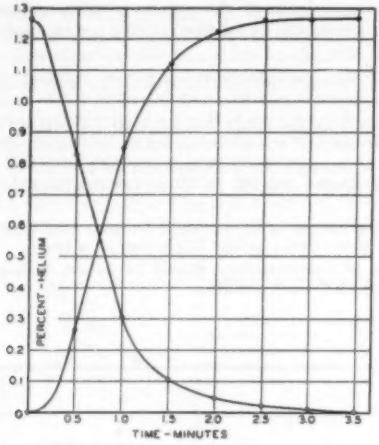

FIG. E—LAG OF KATHAROMETER RESPONSE WITH 3 FT TUBING

The poor agreement between measured and estimated air change rates both at design and average conditions indicates the following conclusions.

1. The statement on page 244 of THE GUIDE 1957 that is very widely used because of its simplicity should be reworded as follows: *An allowance of one-third air change per*

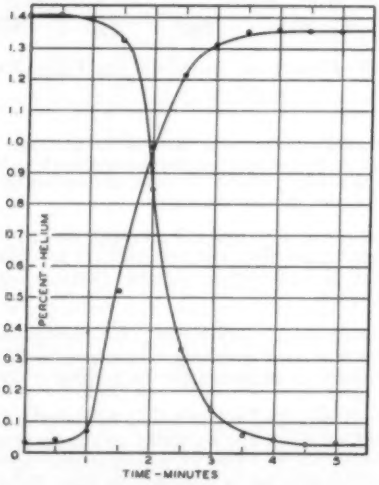

FIG. F—LAG OF KATHAROMETER RESPONSE WITH 30 FT TUBING

hour for all sources of air leakage for the entire volume may be considered average for a well constructed residence at average conditions of wind and temperature difference.

2. All values in Table 4, page 245 of THE GUIDE 1957 should be revised considerably, including the footnote. The title of Table 4 should be reworded as follows: *Air Changes Taking Place Under Design Conditions in Typical Residences, Exclusive of Air Provided for Ventilation.*

3. More measurements of infiltration rate should be made at both design and average conditions of wind and temperature difference.

TABLE D—DESIGN AND AVERAGE RATES OF INFILTRATION FROM THE GUIDE AND FROM THE AUTHORS' PAPERS

ITEM NO.	BASIS OF INFORMATION	DESIGN RATE OF INFILTRATION, AIR CHANGES PER HOUR	AVERAGE RATE OF INFILTRATION, AIR CHANGES PER HOUR
1	1956 Guide, p. 232 1957 Guide, p. 244	Not specified (Obviously more than 1.0)	1.0 (wind velocity and climate not specified)
2	1956 Guide, p. 233 1957 Guide, p. 245	Not specified (Obviously more than 0.69)	0.69 (Text of Part II)
3	1956 Guide, p. 230* 1957 Guide, p. 242	0.75 (at 15 mph)	0.12 (at 5 mph)
4	1956 Guide, p. 230 1957 Guide, p. 242	—	0.40 (at 10 mph)
5	Text of Part II	0.69 (based on Item 2 above)	—
6	Table 6 of Part I	0.61 (at 7 mph, test #5 70 deg. temp. diff.)	—
7	Table 6 of Part I	0.38 (at 14 mph, test #6 54 deg. temp. diff.)	—
8	Table 6 of Part I	—	0.36 (at 10 mph, test #2)
9	Table 6 of Part I	—	0.33 (at 5 mph, test #3)
10	Table 6 of Part I	—	0.22 (at 5 mph, test 12)

* Double-hung wood sash windows, weatherstripped.

4. All measurements should be adjusted for loss of helium by diffusion through walls and ceiling so they can be compared readily with other information.

AUTHORS' CLOSURE (Mr. Harris): Mr. Min's comments relative to the observed wind velocity in proximity to a building as compared to that reported for the area by the weather bureau are interesting. A number of years ago glass surface temperatures were measured on several windows in the Research Home during winter weather. These observations also indicated that the actual wind velocity at the wall of the building seldom exceeded 2 to 5 mph even when weather bureau observations indicated much higher wind movements. On the other hand, it is quite probable that wind effects

would be more pronounced for houses in more exposed locations than that of the Research Home.

In answer to Mr. Min's first question, the abscissa of Fig. 4 is indoor-outdoor static pressure difference measured at the first-story floor level. Thus, it would seem that at an indoor-outdoor pressure difference of 0.000 in. of water the neutral plane should be at 0 in. above the floor of the first story. Fig. 5 shows that an indoor-outdoor pressure difference of 0.000 in. of water represents a day having a low indoor-outdoor temperature difference and low wind velocity. Therefore, it would be expected that the pressure difference between different elevations in the house would be zero also, and the neutral plane could be assumed at any level.

Mr. Min is correct that there should be a minus sign in the equation $Vdc = nV\,dt$. While the minus sign was omitted in this equation, it apparently was considered in the solution of the equation.

Mr. Parsons expresses the opinion that the infiltration rates shown in Table 4, page 245 of the ASHAE GUIDE should be reduced. The authors feel that results obtained in only 2 houses are not sufficient to warrant changes in the material now published in THE GUIDE. They do agree that additional tests of this type are desirable and should be encouraged.

1616

No. 1616

DESIGN AND PERFORMANCE OF A PORTABLE INFILTRATION METER

By CARL W. COBLENTZ* AND PAUL R. ACHENBACH**, WASHINGTON, D. C.

CALCULATION of the heating or cooling load of a structure usually includes a component for the load caused by the infiltration of outside air into the building. The computation of this portion of the heating or cooling load may be based on the length of the cracks around the doors and windows, an estimate of the workmanship involved, the design wind velocity, the degree of exposure to the wind, and the inside-outside temperature difference; or it may be based on a more general estimate of the number of air changes under design conditions. These methods are known to provide only approximations of the true air leakage, but they have been found useful in the absence of an acceptable method for direct measurement of air leakage. A portable infiltration meter was designed and constructed which utilizes the tracer-gas technique to determine the air change rate in different rooms or at different places in a building.

The air change rate of an enclosure is usually defined as the ratio of the hourly rate at which the air enters (or leaves) the enclosure to the volume of the enclosure. The rate of change in concentration of a tracer gas caused by infiltration of outside air can be expressed by the formula:

$$-V\,(dc/dt) = Kc \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where

V = volume of the enclosure.
c = concentration of tracer gas at time t.
K = average volume of air infiltration per unit time for the time interval.

When $c = c_0$ at $t = 0$, the solution of Equation 1 is as follows:

$$c = c_0\, e^{-Kt/V} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

or

$$Kt/V = \log_e (c_0/c) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

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* Mechanical Engineer, U. S. Department of Commerce, National Bureau of Standards. Member of ASHAE.
** Chief, Air Conditioning, Heating, and Refrigeration Section U. S. Department of Commerce, National Bureau of Standards. Member of ASHAE.
Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Murray Bay, Que., Canada, June 1957.

Equation 3 shows that the number of air changes occurring during the time t is equal to the natural logarithm of the ratio of the tracer-gas concentrations at the beginning and at the end of this time interval.

One of the methods for measuring changes in concentration of a tracer gas is to observe the change of thermal conductivity of the mixture of air and the tracer gas. Since the thermal conductivity of helium is about 6 times that of air, small concentrations of helium in air cause a change in the thermal conductivity of the mixture that can readily be measured by means of a thermal conductivity meter. This technique was probably first used by Marley[1] in England for measuring air infiltration of buildings. Other tracer gases and other means of measuring change of concentration of these tracers have been proposed and some are in practical use. Some of these alternate tracer-gas techniques use the following gases and detecting instruments:

1. Ethane gas with an interferometer.
2. Radioactive tracer gas with a sensitive electrometer.
3. Hydrocarbon gas with an infrared gas analyzer.
4. Halogenated hydrocarbon gas with a photometer.
5. Hydrocarbon gas with a sensitive hygrometer to sense changes in the moisture content of samples after combustion of the hydrocarbon.

In designing the portable thermal conductivity meter (Fig. 1), or katharometer, simplification in the apparatus and procedure for making infiltration measurements by the tracer-gas method was sought by applying the following principles:

1. The sensing elements were located in the spaces where the infiltration rate was to be measured, and natural convection moved the mixture of tracer gas and air through them, thus all sensing elements had the same lag in response. The sensing elements were connected to the measuring console with cables.

2. The change in tracer-gas concentration was observed rather than the absolute value since only the change in concentration is required to determine infiltration rate.

3. A saturater was not required for the gas sample since the change of humidity at a given station during a 1-hour infiltration test would usually be small and because absolute values were not being measured.

Although the thermal conductivity of hydrogen is about 7 times that of air, whereas that of helium is only 6 times as high as air, the use of helium is preferable for infiltration measurements because of the explosion hazard of hydrogen. However, when helium is not available, the use of a hydrogen concentration of 0.4 percent does not introduce an acute danger, as the minimum explosive concentration of hydrogen is about 4.1 percent. Hydrogen should not be used when the house is heated with a hot air furnace, since the hydrogen would be slowly oxidized as the hydrogen-laden air was recirculated between the furnace and the living space, and thus introduce a false indication of decay.

DESIGN OF THE SENSING ELEMENTS

The thermal conductivity cells consist of brass blocks 1 × 1¾ × 3¼ in. with two ⅝-in. ID cylindrical cavities, symmetrically arranged. Carefully matched thermistors, of the bead type on a glass probe, are cemented in the cavities with

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[1] The Measurement of the Rate of Air Change, by W. G. Marley (*Journal of the Institution of Heating and Ventilating Engineers*, Vol. 2, 1935).

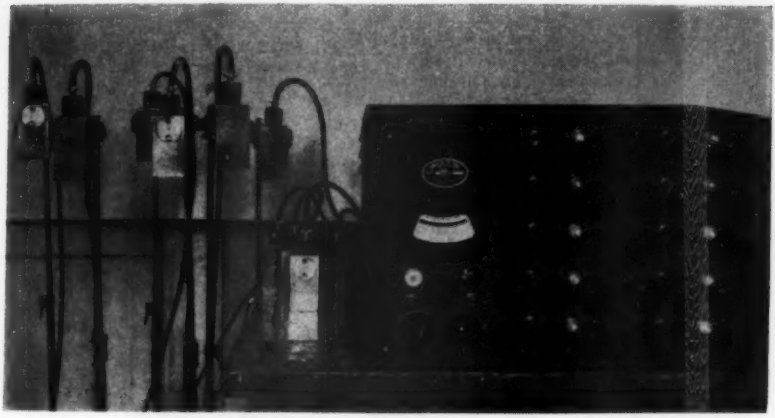

FIG. 1—VIEW OF THE NEWLY DESIGNED PORTABLE INFILTRATION METER

epoxy-type cement. One cavity is hermetically sealed and the other one contains two $\frac{3}{64}$-in. holes each at the top and at the bottom of the cavity to permit natural convection air movement, induced by the heated thermistor. The 2 thermistors are connected in series and are heated with 120 milliwatts to approximately 200 F. This heat input produces a rise in temperature of the brass blocks of less than 1 deg above room temperature. The time constant for the response of the sensing elements was found to be about 76 sec.

Both cavities provide equivalent heat absorbing surfaces. The coefficients of heat transfer by radiation and convection are the same for both cavities, whereas the coefficient of heat transfer by conduction is higher in the cavity which contains the helium-air mixture. Therefore, the thermistor in this cavity will be slightly cooler than the one in the sealed cavity. Small changes in the temperature of the thermistor produce a considerable change of its resistance, such that a change of helium concentration can be determined by observing the difference of the resistance of the 2 thermistors. By installing the 2 thermistors in a Wheatstone bridge circuit, the unbalance produced by presence of helium in 1 cavity can be used as a measure of helium concentration.

DESIGN OF THE MEASURING APPARATUS

The control cabinet housed the bridge circuits and control equipment for 10 sensing elements and was designed for 115-volt a-c operation. Each probe had a polarized outlet and was connected with the cabinet by means of a 3-conductor shielded cable. A schematic wiring diagram of the apparatus showing the power supply and metering circuit and one of the 10 probe circuits is shown in Fig. 2. Two 200-ohm resistors, R_3 and R_5, formed the fixed legs of the Wheatstone bridge for each test probe. Each bridge circuit was balanced with two 10-ohm variable resistors, R_4 and R_6, connected in parallel. One of them, R_4, was connected in series with the fixed 10-ohm resistor, R_7, to provide a coarse and fine adjustment over a rather wide range.

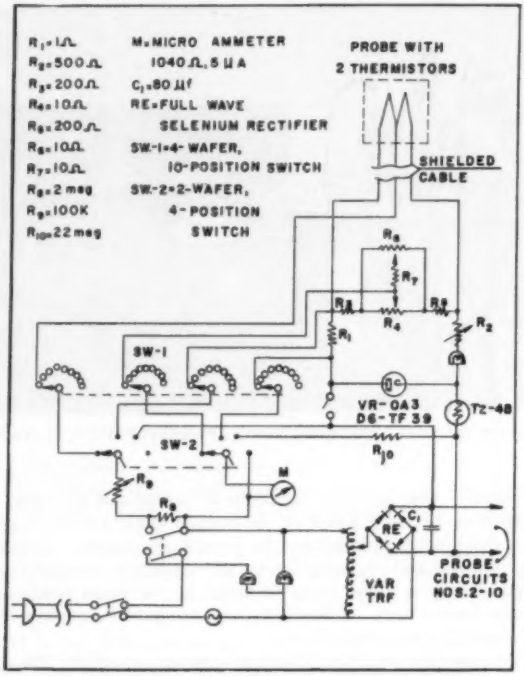

FIG. 2—SCHEMATIC WIRING DIAGRAM OF POWER SUPPLY, THE METERING CIRCUIT AND ONE OF THE PROBE CIRCUITS

The thermistors in each probe were selected to have a minimum difference in resistance over a current range from 12 MA to 22 MA. However, it was found that due to the aging of the thermistors and resistors, and to the accumulation of certain gases produced by the setting of the cement in the sealed cavity, an appreciable balancing range was necessary for the bridge circuit.

A filtered 105-volt direct current supply for each probe circuit was obtained by installing a variable transformer in series with the a-c side of a full-wave selenium rectifier. This variable transformer also compensates for aging of the rectifier and for changes in the output voltage caused by using different numbers of probes.

A constant-wattage network is required for supplying the thermistors. Line voltage fluctuations were reduced by means of a voltage regulator tube, VR-OA3, with a variable resistance tube D6-FT39, as the voltage-dropping resistor. With a proper variable resistor, R_2, and a switchboard bulb, T2-48, operating as pilot light as well as a voltage-dropping resistor, as shown in Fig. 2, fluctuations of the a-c supply voltage of ± 25 percent produce a change in the bridge heater current of less than 0.4 percent. The individual current control for each bridge circuit is necessary because the thermistors require a higher initial voltage to bring them up to temperature.

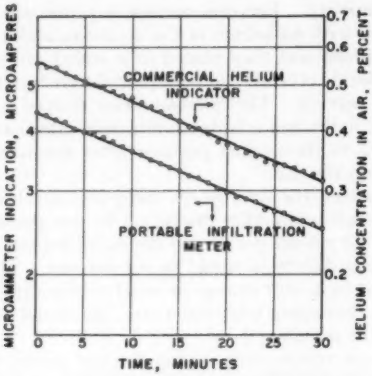

FIG. 3—DECAY CURVES OF PORTABLE INFILTRATION METER AND COMMERCIAL HELIUM INDICATOR

A 1040-ohm microammeter, M, with full scale deflection at 5 micro-ampere current is used as an indicator. Two series-connected selector switches and the necessary shunt and voltage-dropping resistors permit the measurement of the progressive unbalance of the Wheatstone bridge circuit of each probe, the heating current of each probe, the d-c output voltage of the rectifier, and allow shunting of the meter when not in operation. A green pilot light indicates that the line voltage is supplied to the variable transformer and a red pilot light shows that the 2-megohm safety resistor, R_S, in series with the indicating meter, has been shunted.

PERFORMANCE

When air infiltrates into a room at a steady rate, the tracer-gas concentration decreases along a logarithmic curve as shown by Equation 3. Plotted on semi-logarithmic paper the decay curve, then, must be a straight line. A test was made under controlled conditions in the Test Bungalow at the National Bureau of Standards where the outside air movement and inside and outside temperatures were maintained constant. Under such steady conditions the forces producing infiltration should be constant and the rate of air leakage should also be constant. Observations were made on both the portable infiltration meter described herein and a larger commercial helium indicator which employed a sampling system and a pump. The inlet of the sampling line of the commercial instrument was installed adjacent to the probe of the portable meter and the first readings of both instruments were taken at the same time. The observed readings for both instruments are plotted in Fig. 3. The straight lines shown are those best fitted to the observed data based on the least mean squares of the deviations. The straight lines show that both instruments were indicating a logarithmic decay in the helium concentration. The infiltration rate indicated in Fig. 3 by the commercial instrument is 3.23 air changes per hour, whereas that indicated by the portable infiltration meter is 3.19 air changes per hour.

The effect of a change in relative humidity around the sensing probe was determined by measuring the change in the microammeter indication for a known

change in relative humidity. For this purpose a probe was allowed to come to steady-state conditions with no helium in the air for an ambient relative humidity of 28 percent. The probe was then placed in a sealed can into which an ample amount of calcium chloride crystals had been placed to reduce the relative humidity to approximately 0.5 percent. The microammeter showed a change in deflection of 0.20 microamperes, indicating a higher infiltration rate than the true value. It subsequently returned to its original position after removing the desiccant and admitting room air into the can.

On the basis of this test, the error in the computed air-change rate caused by a 1 percent change in relative humidity during a 1-hr test period when the apparent infiltration rate was 1 air change per hr and the initial helium concentration caused a 5.0 microampere scale deflection would be 0.4 percent. The error in computed infiltration rate caused by a unit change in relative humidity should increase with temperature and with decreasing infiltration rate, but would probably be very little affected by the relative humidity level.

A change in the room temperature during the test period would have no effect on the meter reading, if the 2 thermistors in a probe were perfectly matched with respect to their temperature-resistance characteristics over the range of temperature used. Moderate care, but not the ultimate possible, was used in matching the thermistors for the prototype instrument. A test of one of the sensing probes showed that a deflection of 0.35 μA was caused on the indicating meter by a temperature change of 35 deg, *i.e.*, the sensitivity to temperature was 0.01 μA/F.

Under the same test conditions cited for the investigation of sensitivity to relative humidity, the error caused by a 1 deg temperature change would be 0.3 percent. Logically, a standard for matching the thermistors would have to be established if the prototype infiltration meter were put into commercial production.

OPERATIONAL PROCEDURE

For infiltration measurements, the console is placed at a central location in the building and a probe is mounted on a tripod near the middle of each room about 3 ft above the floor. After adjusting the heating current for each probe, at least a ½-hr period should be allowed for all components to warm up to a steady state condition. The bridge circuits are then balanced by adjusting the balancing resistors so that the meter reads zero for each probe used in the test. Helium in the amount of approximately ½ of 1 percent of the total volume of the space is then introduced either directly into the rooms and thoroughly mixed with the room air by using desk-type fans or, where a forced air heating system is in use, the helium is fed into the blower intake and distributed into the rooms through the duct system.

Under some conditions of usage of the infiltration meter, the corrections required to account for changing ambient humidity and temperature during a test could be determined by observing the drift of the microammeter during a rating period preceding the introduction of helium into the space. This method would probably be useful only when the changes in humidity and temperature followed a steady trend during the rating period and the test period.

The apparatus described can be used to study air movement between the different enclosed spaces in a house, such as basement and living quarters, first floor and second floor, living quarters and attic, or between outdoors and any of these spaces. It is sufficiently portable to use in field studies of infiltration in different types of building construction without great inconvenience to the occupants of the building.

1617

No. 1617

PERFORMANCE TESTING OF ROOF VENTILATORS

By J. J. WANNENBURG*, PRETORIA, SOUTH AFRICA

APPLICATION of natural draft roof ventilators to industrial buildings has made apparent the necessity of developing a method whereby the performance of these ventilators may be conveniently tested. The purpose of such tests would be to obtain data to serve (*a*) as a basis for the comparative rating of ventilators, and (*b*) for the prediction of ventilator performance under practical conditions. The investigations, of which the results and conclusions are given here, were directed towards development of a test method which could be applied with expedience to the routine testing of ventilators.

Essentially, 3 factors contribute to the movement of air through a natural draft roof ventilator, viz:

1. Acceleration of air in the vicinity of the ventilator head causes a depression which results in the extraction of air from the building. This is the *aspiration effect.* Its magnitude is a function of ventilator shape, and of windspeed and wind direction at the ventilator.

2. If the outside air is cooler than air inside, the warmer air will move through the ventilator to the cooler outdoor environment. This is the *chimney* or *stack effect.*

3. Generally, in the case of a building exposed to wind, a positive pressure exists over the windward walls, and a negative pressure over the roof and leeward walls[1]. If there are openings in the roof and walls, movement of air results, inward through the windward openings and outward through the leeward and roof openings[2]. Thus there is an additional extraction of air through a roof ventilator, and this flow does not depend upon ventilator shape, but upon wind condition, the resistance to flow of the ventilator, and its position on the roof. In addition, the pressure causing the flow depends on the number and size of other ventilating openings.

This effect is here referred to as the *forced draft effect,* and it must be noted that it is not present during the testing of a ventilator in an open jet wind tunnel, the static

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* Research Officer, National Mechanical Engineering Research Institute, South African Council for Scientific and Industrial Research.

[1] Exponent numerals refer to References.

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pressure at inlet to the stack extension (or suction pipe) being, under these conditions, equal to that in the free windstream. It is clear, therefore, that the results of such a ventilator test give information pertaining only to the aspiration effect.

It was decided that the comparative smallness of the stack effect, and the dependence of the *forced draft effect* on such irreproducible factors as the structural features of the building, would not justify separate experimental evaluation of these effects, and a method is therefore given for estimating their magnitudes theoretically.

It would be difficult to calculate accurately the extraction of air through a ventilator without prior knowledge of the windspeed and wind direction at the actual location of a ventilator, and the static pressure at the stack inlet. Such prior knowledge implies either tests on the full scale building, or wind tunnel tests on a scale model. However, if the accuracy of the required information regarding the ventilator performance does not justify such procedures, a rough estimate may be obtained by assigning empirical or average values to the unknown factors.

It might be argued that, if test results can be applied only to obtain a rough estimate of ventilator performance, there is no point in seeking accuracy in these results. However, there is a purpose for which accurate test results are essential, and that is for the

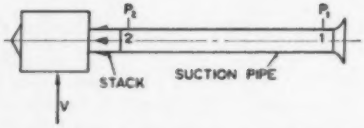


FIG. 1—DIAGRAMMATIC REPRESENTATION OF VENTILATOR AND SUCTION PIPE

comparative rating of different types of ventilator. In this connection results obtained from tests in an open jet wind tunnel provide a convenient basis for comparison. As mentioned before, such results apply to the aspiration effect, and this effect, all other factors being equal, is dependent on ventilator shape and resistance.

THEORETICAL CONSIDERATIONS

Aspiration Effect and Free Inlet Correction: Testing of a roof ventilator necessitates use of a duct attached to the ventilator stack, in which the induced volumetric flow may be measured. This duct, which will be referred to as the *suction pipe*, increases resistance to air flow of the system, and since in practice a ventilator usually operates without any such extension (*i.e.* under *free inlet conditions*) a method had to be devised whereby test results could be corrected or extrapolated to free inlet conditions. To this end the following method was proposed by Schrenk[3,4].

Fig. 1 represents a ventilator attached to a suction pipe, the ventilator projecting into a windstream of velocity V fps. Let the resulting velocity of flow in the suction pipe be v_w fps, and let the static pressures as measured at positions (1) and (2) in the suction pipe be P_1 and P_2 respectively. Then the static depressions at (1) and (2) may be expressed as

$$\Delta P_1 = (P_a - P_1) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

$$\Delta P_2 = (P_a - P_2) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where

P_a = atmospheric pressure.

Schrenk then introduced the nondimensional parameters ψ and η, where

$$\psi = v_w/V = \text{flow coefficient} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3a)$$

and

$$\eta = (\Delta P_2 - \Delta P_1)/\tfrac{1}{2}\rho_0 V^2 = \text{pressure coefficient} \quad . \quad . \quad . \quad . \quad . \quad (3b)$$

ρ_0 = air density at wind temperature and the velocity of the free windstream.

Schrenk postulated that, if the orientation of the ventilator in the wind remains unchanged, then for all values of suction pipe resistance, the relation between ψ and η will be given by

$$\psi = \beta\,(\eta_{max} - \eta) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

where

η_{max} = value of η for a suction pipe of infinite resistance, *i.e.* when the duct is closed off (in which case $\Delta P_1 = 0$).

Equation 4 may be written in a slightly different form by introducing the parameter λ, where

$$\lambda^2 = \Delta P_1/\tfrac{1}{2}\,\rho_0 V^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Now, according to Bernoulli

$$\Delta P_1 = P_a - P_1 = (1 + K_1)\,\tfrac{1}{2}\rho_i v_w^2$$

where

K_1 = resistance of that portion of the suction pipe upstream of the point (1)
v_w = airspeed in the pipe.
ρ_i = average density of the air in the suction pipe.

Therefore

$$\lambda^2 = \frac{\Delta P_1}{\tfrac{1}{2}\rho_0 V^2} = (1 + K_1)\left(\frac{v_w^2}{V^2}\right)\left(\frac{\rho_i}{\rho_0}\right) = (1 + K_1)\left(\frac{\rho_i}{\rho_0}\right)\psi^2 \quad . \quad . \quad . \quad (6)$$

Hence, provided K_1 is constant and the densities remain unchanged, λ will be directly proportional to ψ. Schrenk's equation may now be written

$$\lambda = \alpha(\eta_{max} - \eta) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

$$\alpha = \sqrt{(1 + K_1)}\,\beta \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

Equation 7 represents a straight line as shown in Fig. 2.

Thus, if 2 points A and B on this line are obtained experimentally, then the value of λ at $\eta = 0$ may be read off, this value corresponding to free inlet conditions (*i.e.* where $\Delta P_1 = \Delta P_2$). Substituting this value of λ (*i.e.* λ_0) into Equation 7 would yield the flow coefficient at free inlet conditions. It was implicitly assumed that the parameter ψ remained constant and independent of the windspeed, for a constant value of the suction pipe resistance; in other words, that the airspeed in the suction pipe was directly proportional to the windspeed, the factor of proportionality (ψ) being dependent on the resistance. This assumption was experimentally confirmed at a later stage in these investigations.

One aspect of Schrenk's method which renders its validity dubious, is the assumption implied in Equation 4, that a velocity should be directly proportional to a pressure, experience indicating that, for turbulent flow, pressures are proportional to the square of a velocity. On this account it was decided to obtain a theoretical relationship between the parameters employed by Schrenk. This relation would then be compared with Schrenk's empirical equation and would subsequently be subjected to experimental verification.

The arguments employed in this connection were based on the following assumptions:

1. For a particular orientation of the ventilator with respect to the wind, the dimensionless parameters ψ and $\Delta P/\frac{1}{2}\rho_0 V^2$ are constant for any value of V larger than about

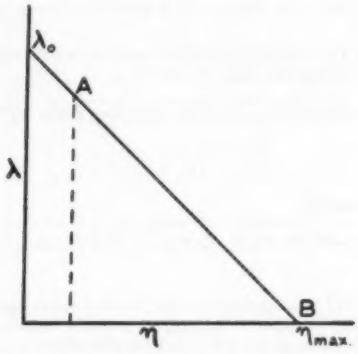


FIG. 2—SCHRENK'S LINE

3 fps, where ΔP is any depression as measured in the suction pipe, the lower limit of V being introduced to eliminate the possibility of non-turbulent flow.

2. The second assumption involves the static depression $\Delta P_3 = (P_a - P_3)$ which causes the flow through the suction pipe and ventilator, this depression being defined as

$$\Delta P_3 = (1 + K + K_v)\ \tfrac{1}{2}\rho_1 v_w^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

where

K = total suction pipe resistance.
K_v = ventilator resistance.
$\frac{1}{2}\rho_1 v_w^2$ = mean velocity head in suction pipe.

The depression ΔP_3 exists within the ventilator but, due to the intricate internal construction of the ventilator and the interference of the wind, it cannot, in general, be measured directly. There is, however, one condition where direct measurement is possible, and that is when the entrance to the suction pipe is closed off. Under these conditions it may be assumed that the depression ΔP_{inf} which exists within the ventilator also exists within the suction pipe. If now the resistance of the suction pipe is decreased from infinity to a value K, then there will be a certain

amount of flow through the ventilator and the depression within the ventilator will change from ΔP_{inf} to ΔP_3. Assume now that

$$\Delta P_3 = \Delta P_{inf} - x \tfrac{1}{2} \rho_i v_w^2 \quad . . . . . (10)$$

where

x = a parameter depending only upon the relative angle between the wind and the ventilator.

Keeping Equation 9 in mind, it will be clear that the equivalent system representing the ventilation and suction pipe will be as given in Fig. 3. Here K_i is the inlet resistance, K_p is the suction pipe resistance between points (1) and (2) and K_v is the ventilator resistance. The pressure at (2) will be given by

$$\Delta P_2 - \Delta P_1 = K_p \tfrac{1}{2} \rho_i v_w^2 \quad . . . . . (11)$$

Similarly

$$\Delta P_3 - \Delta P_2 = K_v \tfrac{1}{2} \rho_i v_w^2 \quad . . . . . (12)$$

where

ΔP_1, ΔP_2 and ΔP_3 = depressions at points (1), (2) and (3) respectively

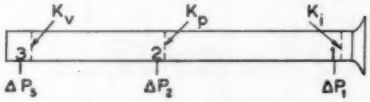


FIG. 3—EQUIVALENT SYSTEM FOR VENTILATOR AND SUCTION PIPE

Dividing Equation 12 by the wind velocity head $\frac{1}{2}\rho_0 V^2$, gives:

$$\frac{\Delta P_3}{\frac{1}{2}\rho_0 V^2} - \frac{\Delta P_1}{\frac{1}{2}\rho_0 V^2} = \eta + K_v \psi^2 \frac{\rho_i}{\rho_0} \quad . . . . . (13)$$

Substituting from Equations 10 and 6, Equation 13 becomes

$$\frac{\Delta P_{inf}}{\frac{1}{2}\rho_0 V^2} - x \psi^2 \frac{\rho_i}{\rho_0} = \eta + K_v \psi^2 \frac{\rho_i}{\rho_0} + \lambda^2 \quad . . . . . (14)$$

Now $\Delta P_{inf}/\frac{1}{2}\rho_0 V^2$ represents the maximum value of η, i.e. the value of η when the suction pipe is closed off (infinite resistance). If this value is written as η_{max}, then

$$\eta_{max} = \Delta P_{inf}/\tfrac{1}{2} \rho_0 V^2 \quad . . . . . (15)$$

Substituting for λ^2 from Equation 6, Equation 14 becomes:

$$\eta_{max} - \eta = \lambda^2 \left(1 + \frac{x + K_v}{1 + K_i}\right) \quad . . . . . (16)$$

or

$$\lambda^2 = \alpha (\eta_{max} - \eta) \quad . . . . . (17)$$

where

α = a constant depending only upon the relative angle between the wind and the ventilator.

Comparing this equation with Equation 7 it is seen to differ from that of Schrenk in that the relation between λ and η is parabolic instead of linear, a relation which would satisfy the requirements of turbulent flow more satisfactorily. The graphical representation of Equation 17 would be the same as that depicted in Fig. 2 for Schrenk's relation, providing the ordinate axis is taken as λ^2 instead of λ. Hence the remarks concerning the method of correcting to free inlet conditions as regards Schrenk's equation also applies to Equation 17. However, since in Equation 17 the value of the constant α is known in terms of quantities which are separately measurable, this equation allows for the calculation of the performance at any value of the suction pipe resistance.

In this connection, consider Equation 11. Dividing by $\frac{1}{2}\rho_0 V^2$ and substituting for η gives:

$$\eta = K_p\,(\tfrac{1}{2}\rho_i v_w^2/\tfrac{1}{2}\rho_0 V^2) = (\rho_i/\rho_0)\,K_p\psi^2$$

Substituting this expression, together with Equation 6 into Equation 16 yields:

$$\psi^2 = [\eta_{max}\,(\rho_0/\rho_i)]/[1 + x + K_i + K_p + K_v]$$

and it follows, since the density is inversely proportional to the absolute temperature, that

$$\psi^2 = \frac{\eta_{max}\,(T_i/T_0)}{1 + x + K_i + K_p + K_v} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

where

T_0 = temperature of air (absolute scale) outside the building.
T_i = temperature of air in the suction pipe.

If $K_i + K_p$, the total suction pipe resistance, is denoted by K, Equation 18 becomes

$$\psi^2 = [\eta_{max}\,(T_i/T_0)]/[1 + x + K + K_v] \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

By performing tests (one with an open and another with a closed suction pipe), the value of the constant α in Equation 17 may be found, this constant being equal to

$$\alpha = 1/[1 + (x + K_v)/(1 + K_i)] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

K_i and K_v may be determined separately, so that x may be found. Substituting x and K_v into Equation 19 and assuming, as is the case under test conditions, that $T_0 = T_i$, a relation between the flow coefficient ψ and the suction pipe resistance K is obtained, from which the airspeed in the suction pipe can be calculated for any value of K and V. In particular, for free inlet conditions $K = 0$ and

$$\psi_0^2 = [\eta_{max}\,(T_i/T_0)]/[1 + x + K_v] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (21)$$

where

ψ_0 is the flow coefficient for free inlet conditions.

Since $\psi = v_w/V$, the air velocity in the suction pipe is given by

$$v_w = V\sqrt{[\eta_{max}(T_i/T_0)]/[1 + x + K + K_v]} \quad . \quad . \quad . \quad . \quad . \quad (22)$$

The extraction, in cubic feet per minute of air at a temperature T_i, is given by

$$Q_w = 88AVC_t\sqrt{\frac{\eta_{max}}{(1 + x + K + K_v)}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (23)$$

where

V = miles per hour.
A = cross-sectional area of ventilator stack (assumed same as that of suction pipe).
C_t = temperature correction factor = $\sqrt{T_i/T_0}$.

Defining standard air conditions as dry air at a temperature of 70 F and a baro-

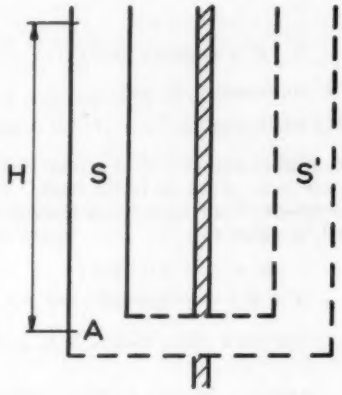


FIG. 4—DIAGRAM ILLUSTRATING STACK EFFECT

metric pressure of 760 mm mercury, the extraction in cubic feet per minute of standard air is given by

$$Q_{w,standard} = 88AVC_tC_s\sqrt{\frac{\eta_{max}}{(1 + x + K + K_v)}} \quad . \quad . \quad . \quad . \quad (24)$$

where

$$C_s = (530/760)\ (B/T_i) = 0.697\ (B/T_i) \quad . \quad . \quad . \quad . \quad . \quad . \quad (25)$$

B being the barometric pressure in millimeters mercury.

For ordinary purposes the temperature correcting factor C_t may usually be taken as unity.

Stack Effect: A formula for estimating the magnitude of the stack effect will now be derived. Consider a stack of height H (ft) in which the air temperature has the average value of T_i in F absolute, and let S' be an identical column of air outside the building in which the average air temperature is T_0, in F absolute, (see Fig. 4).

If T_i is greater than T_0, the column S' will, provided S and S' are interconnected at the bottom, cause the air in S to move upwards. The pressure at A causing this movement is due to the difference in weight between the 2 columns, and will be equal to

$$h_t = H\,(W_0 - W_i) \text{ lb per sq ft} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (26)$$

where

W_0 = density of air, pounds per cubic foot at temperature T_0.
W_i = density of air, pounds per cubic foot at temperature T_i.

Now the density of air, in slugs per cubic foot at a temperature T and a barometric pressure B (mm mercury) is given by

$$\rho = 0.001625\ B/T$$

so that

$$W = 0.001625\ (B/T)_g$$

Substitute for W_0 and W_i in Equation 26, and

$$h_t = H \times 0.001625gB[(1/T_0) - (1/T_i)] \text{ lb per sq ft} \quad . \quad . \quad . \quad . \quad (27)$$

Let v_s be the air velocity (fps) in the stack, K_t the total stack resistance factor and ρ_i the density, in slugs per cu ft, of the air in the stack. Then the pressure head necessary to set up the velocity head $\frac{1}{2}\rho_i v_s^2$ plus that required to overcome the friction head $K_t\ \frac{1}{2}\rho_i v_s^2$, is equal to

$$h_s = (1 + K_t)\ \tfrac{1}{2}\rho_i v_s^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (28a)$$

$$= (1 + K_t)\ \tfrac{1}{2} \times 0.001625\ (B/T_i)v_s^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad (28)$$

If flow is provided only by the stack effect, then $h_t = h_s$, and it follows from Equations 27 and 28 that

$$v_s = [2Hg/(1 + K_t)]^{1/2}\,[(T_i/T_0) - 1]^{1/2}, \text{ fps}$$

The flow Q_s, in cfm of air at temperature T_i is given by

$$Q_s = 60A\sqrt{\frac{2Hg}{(1 + K_t)}}\left[\frac{T_i}{T_0} - 1\right] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (29)$$

where

A = the area, in square feet, of the suction pipe.

Substituting for g:

$$Q_s = 482A\,[H/(1 + K_t)]^{1/2}\,[(T_i/T_0) - 1]^{1/2}$$

and since

$$K_t = K + K_v$$

$$Q_s = 482A\sqrt{\frac{H}{(1 + K + K_v)}}\left[\frac{T_i}{T_0} - 1\right]^{1/2} \quad . \quad . \quad . \quad . \quad . \quad . \quad (30)$$

The extraction, in cfm of standard air, is given by

$$Q_{w,\text{standard}} = 482AC_s\sqrt{\frac{H}{(1 + K + K_v)}}\left[\frac{T_i}{T_0} - 1\right]^{1/2} \quad . \quad . \quad . \quad (31)$$

Forced Draft Effect: As mentioned earlier, this effect depends upon the existence of a pressure difference between the interior of a building and the region over the roof. If this pressure difference h_f is known, the extraction due to the forced draft may readily be calculated from a knowledge of the ventilator resistance. Thus

$$h_f = (1 + K + K_v)\tfrac{1}{2}\rho_i v_f^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (32)$$

where

$\tfrac{1}{2}\rho_i v_f^2$ = average velocity head in suction pipe.

Substituting for ρ_i in terms of the temperature T_i and the barometric pressure B, and expressing h_f in mm water, the extraction in cfm of air at temperature T_i is given by

$$Q_f = 0.952 \sqrt{\frac{h_f}{(1 + K + K_v)}} \left[\frac{T_i}{B}\right]^{1/2} \quad . \quad . \quad . \quad . \quad . \quad . \quad (33)$$

The extraction in cfm of standard air is equal to

$$Q_{f(\text{standard})} = 0.952 C_s \sqrt{\frac{h_f}{(1 + K + K_v)}} \left[\frac{T_i}{B}\right]^{1/2} \quad . \quad . \quad . \quad . \quad (34)$$

Combined Effects of Wind and Temperature: To calculate total flow when the stack effect acts in combination with the aspiration and forced draft effects, consider Equation 10.

$$i.e.\ \Delta P_3 = \Delta P_{inf} - x\tfrac{1}{2}\rho_i v_w^2$$

since

$\eta_{max} = \Delta P_{inf}/\tfrac{1}{2}\rho_0 V^2$ and $\psi^2 = (v_w^2/V^2)$, it follows that

$$\Delta P_3 = (\eta_{max} - x\psi^2)\,\tfrac{1}{2}\rho_0 V^2$$

$$= \left[\eta_{max} - \frac{x\eta_{max}}{(1 + x + K + K_v)}\right](\tfrac{1}{2}\rho_0 V^2), \text{ from Equation 19.}$$

When an additional head $h_s + h_f$ is set up, then the depression causing the flow is $\Delta P'_3$, where

$$\Delta P'_3 = \Delta P_3 + h_s + h_f$$

and the total velocity of flow v_{tot}, is given by

$$\tfrac{1}{2}\rho_i v_{tot}^2 = \frac{\Delta P'_3}{1 + K + K_v}$$

$$= \frac{\eta_{max}}{1 + x + K + K_v}\left[\tfrac{1}{2}\rho_0 V^2 + \frac{h_s}{1 + K + K_v} + \frac{h_f}{1 + K + K_v}\right]$$

$$v_{tot}^2 = \rho_0/\rho_i\,[\eta_{max}/(1 + x + K + K_v)]V^2 + [2h_s/\rho_i(1 + K + K_v)]$$

$$+ [2h_f/\rho_i(1 + K + K_v)] = v_w^2 + v_s^2 + v_f^2$$

from Equations 19, 28*a* and 32. Therefore

$$Q_{tot} = \sqrt{Q_w^2 + Q_s^2 + Q_f^2}, \text{ cfm of air at temperature } T_i \quad . \quad . \quad . \quad (36)$$

The total extraction in cfm of standard air is given by

$$Q_{\text{tot (standard)}} = C_s\sqrt{Q_w^2 + Q_s^2 + Q_f^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (37)$$

where Q_w, Q_s, Q_f are given respectively by Equations 23, 30 and 33.

If, as has been found in the case of low-pitched roofs[1], the ventilator is situated within a stagnant zone (*i.e.* a region of no flow), then the aspiration effect is absent, and the forced draft head, h_f, is equal to the difference in static pressure in-

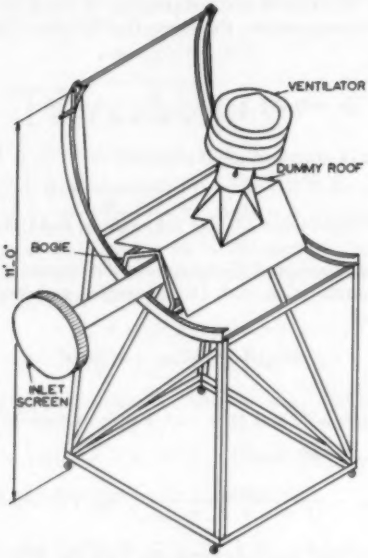


FIG. 5—TEST RIG FOR MOUNTING ROOF VENTILATOR IN WIND TUNNEL

side the building, and that prevailing within the stagnant zone. As mentioned before, these pressures depend upon the building shape and orientation, and must be determined by field or model tests.

EXPERIMENTAL EQUIPMENT

To prove the validity of the theory underlying the method of correcting to free inlet conditions, tests were performed on a typical 12-in. roof ventilator, the tests being conducted in a 10-ft open jet wind tunnel in which the airspeed was variable in the range 5 fps to 30 fps. To provide for testing at different values of the relative angle between the wind and the ventilator, a test rig was designed (see Fig.

5) which supported the ventilator at the correct height in the tunnel working section, and which allowed for tilting the ventilator at various angles to the horizontal. The ventilator, together with its suction pipe, was attached to a bogie moving on 2 parallel curved rails, allowing the relative angle between wind and ventilator to be varied from 0 to 90 deg, this range being extended a further 90 deg by rotating the entire rig horizontally through 180 deg. Adjustment of the bogie position was obtained by a manually operated hoist arrangement which could be locked by a pawl and ratchet.

To approach the manner in which roof ventilators are mounted in practice as closely as possible, a dummy roof (5 ft square when flat) of variable pitch was provided between the bogie and the ventilator stack. The pitch of the dummy roof was, in this case, fixed at 20 deg to accommodate the ventilator stack adaptor.

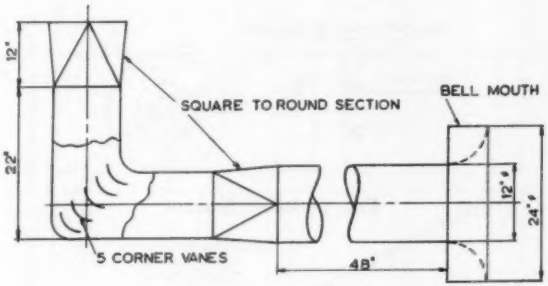


FIG. 6—SUCTION PIPE

The suction pipe consisted of a circular duct connected through a 90 deg elbow to the ventilator stack (see Fig. 6), the elbow being of square section and incorporating 5 sheet metal corner vanes. To provide smooth entrance conditions, the suction pipe inlet was provided with a suitable bell mouth.

Experience with previous tests in the wind tunnel indicated that random disturbance effects could be expected in the *still air* region of the working section, and it was feared that such disturbances acting in the vicinity of the suction pipe inlet would result in a ventilator action opposing the main flow. To avoid this, the inlet to the suction pipe was provided with a screen, and the most suitable design is depicted in Fig. 7. It consisted essentially of 2 concentric rings of resistance material fitted co-axially to the end of the bell mouth, the open ends being closed off by means of hardboard discs. Thus it was expected that any disturbance penetrating through the first ring, would be damped out in the annulus between the 2 rings. A satisfactory resistance material was found to be perforated sheeting, the perforations being of $\frac{1}{8}$-in. diameter at a $\frac{3}{16}$-in. pitch.

Static tappings were provided in the suction pipe at positions immediately inside the inlet and downstream of the elbow, these positions corresponding respectively

to (1) and (2) in Fig. 1. The tappings consisted of 1 mm holes spaced radially at equal distances round the circumference of the duct.

VERIFICATION OF THEORY

The first assumption to be tested was the one postulating that static pressures in the suction pipe are proportional to the wind dynamic pressure, this being equivalent, in effect to the statement that the airspeed in the suction pipe is directly proportional to the windspeed. Measurement of ΔP_1, and ΔP_2, the static depressions at positions (1) and (2) (Fig. 1), were made at different values of the windspeed, the windspeed being measured by Pitot-static tube and all pressure differences by projection manometers. The tests were repeated for values of Θ, the wind incidence, between -20 deg and 53 deg, the sign convention being chosen so that an incidence of $+90$ deg refers to a wind blowing vertically down into the ventilator

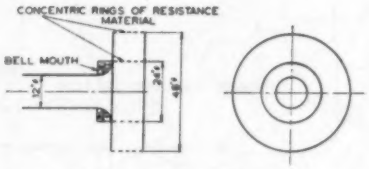


FIG. 7—INLET SCREEN

stack. The results are plotted in Figs. 8, 9 and 10, and the relation was consistently linear. Thus the assumption was justified and the parameters $\Delta P_1/\frac{1}{2}\rho_0 V^2$ and $\Delta P_2/\frac{1}{2}\rho_0 V^2$ could be considered as constant and dependent only upon the ventilator orientation and suction pipe resistance. It must, however, be specified that the constancy of these parameters cannot be guaranteed for values of the windspeed lower than 2 to 3 mph., in which range the flow might no longer be fully turbulent.

The validity of the theoretically derived equation

$$\lambda^2 = \alpha\,(\eta_{max} - \eta)$$

as opposed to Schrenks relation, which has the form

$$\lambda = \alpha\,(\eta_{max} - \eta)$$

was tested by determining the values of λ, η and η_{max} at different values of the suction pipe resistance, η_{max} being obtained by closing off the entrance to the suction pipe. The resistance was varied by introducing wire gauzes of different mesh into the suction pipe.

The results are plotted in Figs. 11, 12 and 13. The values of λ, η and η_{max} were calculated from the slopes of these lines, and the curves of λ against η are plotted for $\Theta = 0$ deg in Fig. 14, and for $\Theta = 18$ deg in Fig. 15. These curves indicate that the theoretical form of the equation agrees better with the experimental results than does Schrenk's version, and it was concluded that more accurate free inlet

corrections would be obtained by the use of the theoretical equation. This equation allows for the prediction of the performance at any value of the suction pipe resistance, thus making it possible to calculate the extraction for the case where an extension is attached to the ventilator stack.

PERFORMANCE TESTING OF A TYPICAL ROOF VENTILATOR

To ascertain the practibility of the proposed test method a trial test was carried out on a typical 12-in. roof ventilator.

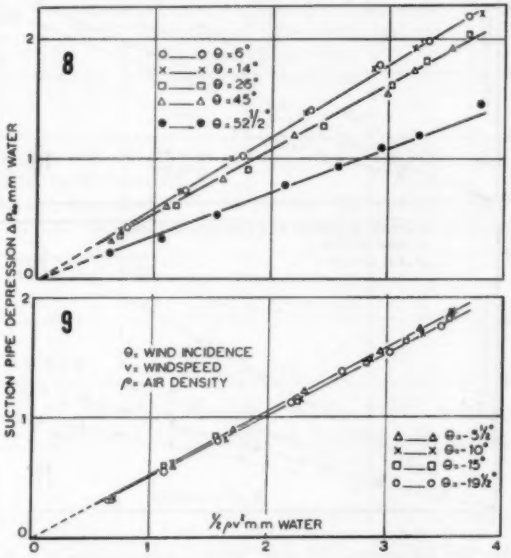


FIG. 8 (TOP) AND FIG. 9 (BELOW)—SUCTION PIPE DEPRESSION FOR INFINITE SUCTION PIPE RESISTANCE

If it is required only to evaluate the aspiration effect it would suffice to determine experimentally the value of the inlet resistance K_1 (which amounts to a calibration of the suction pipe depression in terms of the airspeed), and the values of the parameters η_{max} and α over a range of wind directions. Thus, since K_1 is known, the values of $(x + K_v)$ over the range of wind directions may be found from Equation 20, which, on substitution in Equation 19 yields the corresponding values of ψ. It is clear, therefore, that x and K_v need not be known separately.

However, if it is also required to calculate the stack and forced draft effects, then K_v has to be known explicitly (see Equations 30 and 33). Thus to obtain all the ventilator characteristics, the following experimental procedure is neces-

sary: (*a*) Determination of K_i, the inlet resistance; (*b*) Determination of K_v, the ventilator resistance; and (*c*) Determination of λ, η and η_{max} over a range of wind directions.

If interest is restricted to the aspiration effect (for instance when ventilators are merely to be compared), then step (*b*) is eliminated.

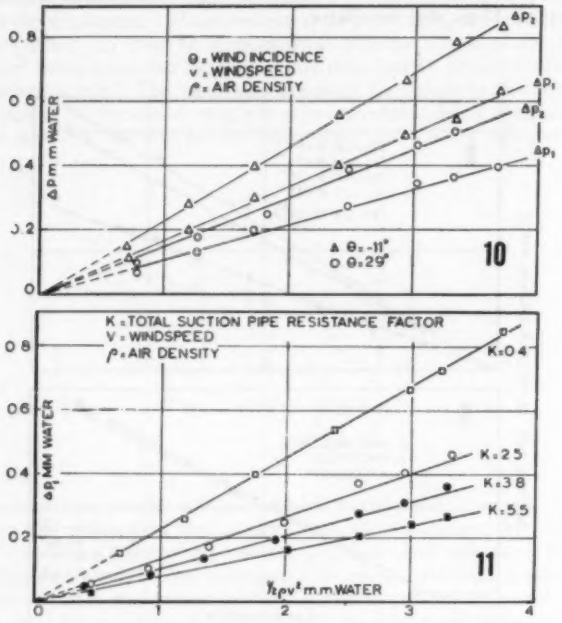


FIG. 10 (ABOVE)—SUCTION PIPE DEPRESSION—NO ADDITIONAL RESISTANCE IN SUCTION PIPE
FIG. 11 (BELOW)—SUCTION PIPE DEPRESSION ΔP_1 FOR DIFFERENT VALUES OF THE SUCTION PIPE RESISTANCE. RELATIVE ANGLE BETWEEN WIND AND VENTILATOR = 0 DEG

(*a*) *Inlet Resistance*: The set-up by means of which K_i was determined is shown in Fig. 16. The 7-deg diffusor was provided with splitters to minimize the effect of possible swirl. Control of the airspeed was obtained by mechanically adjusting the speed ratio between motor and fan, and by introducing various resistances on the outlet side of the fan, this arrangement allowing the airspeed in the suction pipe to be variable within the range zero to 17 fps. ΔP_1 was measured at a position 2-in. inside the inlet to the suction pipe, the use of 4 symmetrically spaced

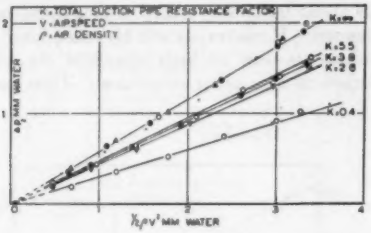


FIG. 12—SUCTION PIPE DEPRESSION ΔP_2 FOR DIFFERENT VALUES OF THE SUCTION PIPE RESISTANCE. RELATIVE ANGLE BETWEEN WIND AND VENTILATOR = 0 DEG

tappings in the duct wall allowing an average value of the static depression at the section of the duct to be obtained.

The graph of ΔP_1 against $\frac{1}{2}\rho V^2$, the suction pipe velocity head, obtained by systematically varying the airspeed, is plotted in Fig. 17, and is seen to be a straight line.

Now, according to Bernoulli

$$\Delta P_1 = (1 + K_1)\tfrac{1}{2}\rho V^2$$

so that

$$K_1 = (\Delta p_1/\tfrac{1}{2}\rho V^2) - 1$$

Therefore, since $\Delta P_1/\frac{1}{2}\rho V^2$, as obtained from Fig. 17, is constant, K_1 must be constant. Thus it was found that an airspeed of about 2 fps in the suction pipe

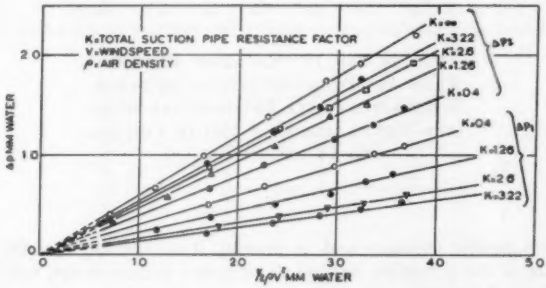
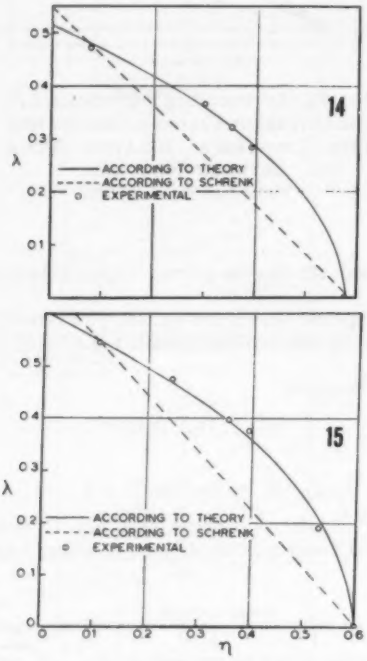


FIG. 13—SUCTION PIPE DEPRESSIONS ΔP_1 AND ΔP_2 FOR DIFFERENT VALUES OF THE SUCTION PIPE RESISTANCE. RELATIVE ANGLE BETWEEN WIND AND VENTILATOR = 18 DEG

yielded a depression of about 0.022 mm water, which could be measured to within 10 percent by a manometer. However, as will be noted later, the testing of a ventilator requires measurements only at high values of the windspeed, the theory allowing for extrapolation to any other condition. Thus measurements may be



FIGS. 14 AND 15—RELATION BETWEEN FLOW AND PRESSURE COEFFICIENTS FOR A RELATIVE ANGLE BETWEEN THE WIND AND VENTILATOR OF 0 DEG IN FIG. 14; AND 18 DEG IN FIG. 15

obtained with greater accuracy and, in general, it is expected that the results obtained by use of the foregoing method of air-speed measurement, will be accurate to within about 5 percent.

(*b*) *Ventilator Resistance*: The apparatus used for obtaining the ventilator resistance is shown diagrammatically in Fig. 18, and consisted of a 12-in. duct connecting the outlet of a 12-in. axial flow fan to the ventilator stack. The flow was measured by the standard orifice plate, and a ring of 4 symmetrically spaced static

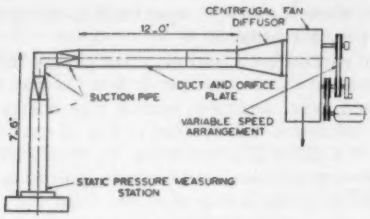


FIG. 16—APPARATUS FOR DETERMINING SUCTION PIPE INLET RESISTANCE (K_1)

tappings as shown. This allowed the pressure loss ΔP over the ventilator to be measured in terms of the airspeed, v. K_v was calculated from the relation.

$$\Delta P = (K_v - 1)\tfrac{1}{2}\rho v^2$$

(*c*) *Performance Tests*: The performance tests amounted to finding the values of the parameters λ, η and η_{max} at different values of the wind direction, using the experimental equipment just described.

By virtue of the constancy of the parameters $\Delta P_1/\tfrac{1}{2}\rho_0 V^2$, $\Delta P_2/\tfrac{1}{2}\rho_0 V^2$ and $\Delta P_{inf}/\tfrac{1}{2}\rho_0 V^2$, it was sufficient to take measurements at only one value of the windspeed, this speed being chosen high enough (about 25 fps) to provide for accurate measurements of pressures. To ensure accuracy each pressure measurement was taken as the average of 10 readings, the observations being repeated, in doubtful cases, at different values of the windspeed. The tests were performed over a range of wind directions from -20 to 67 deg, the higher value representing that angle where the flow through the suction pipe was reversed. Thus, the parameters $\Delta P_1/\tfrac{1}{2}\rho_0 V^2$ and $\Delta P_2/\tfrac{1}{2}\rho_0 V^2$ and $\Delta P_{inf}/\tfrac{1}{2}\rho_0 V^2$ were obtained, and are plotted against Θ in Fig. 19.

CALCULATIONS AND PRESENTATION OF RESULTS

From Fig. 19 the parameters λ, η and η_{max} could be determined which, on substitution in Equation 17 allowed the values of α to be calculated over the range of

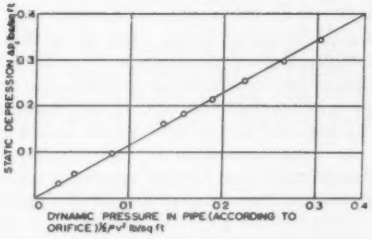


FIG. 17—CALIBRATION OF STATIC DEPRESSION ΔP_1, IN TERMS OF AIRSPEED IN SUCTION PIPE

values of Θ. Further, since K_v and K_i were known, the corresponding values of x could be calculated from Equation 20.

The 2 characteristic parameters x and η_{max} are plotted in Fig. 20, from which, on substitution in Equation 23, the volumetric flow Q_w may be calculated for any values of the windspeed, wind direction, suction pipe resistance and temperature ratio. The results as calculated with the use of Fig. 20 may be more conveniently presented in the form of a chart, as given in Fig. 21, from which Q_w may be directly read off for any required conditions. An example indicating the use of this chart is given on Fig. 21. The extraction as obtained from the chart is given for the conditions where $T_1 = T_0$. If there should be a temperature difference, Q has to be multiplied by the temperature correction factor C_t. In order to obtain the extraction in cfm of standard air, Q has to be multiplied by C_tC_s (see Equation 25).

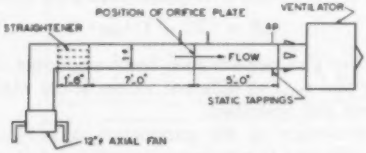


FIG. 18—APPARATUS FOR DETERMINING VENTILATOR RESISTANCE

The stack effect and forced draft effect may be calculated by the use of Equations 30 and 33 respectively, and the combined flow by means of Equation 36.

COMPARATIVE RATING OF VENTILATORS

As mentioned before, the aspiration effect, being the only effect which depends primarily upon the shape and construction of the ventilator, constitutes a convenient basis for comparative rating of ventilators. Since the aspiration effect is characterized by the flow coefficient ψ, and since from Fig. 21, the flow coefficient varies considerably with the wind direction, it is suggested that an average value of ψ (at free inlet) be employed for the purpose of comparison. The range of Θ over which ψ is averaged should, however, be the same for all the ventilators which are to be compared. Averaging ψ over the range from -20 to 67 deg a value of approximately 0.4 is obtained for the ventilator of which the performance curves are given in Fig. 21. This number might be termed the characteristic flow coefficient in the range -20 to 67 deg, and is the value which should be employed when information regarding the wind conditions at the ventilator head is inadequate. The corresponding average extraction for this ventilator under free inlet conditions amounts to about 28 cfm per unit velocity (mph) of the free windstream.

CONCLUSIONS

1. The theory here developed yields closer agreement with the experimental results than does Schrenk's relation. In addition to allowing for the extrapolation of test

results to free inlet conditions, this theory provides a method whereby the performance may be calculated at any given value of the suction pipe resistance, and which allows for the estimation of the extraction when an extension is made to the ventilator stack.

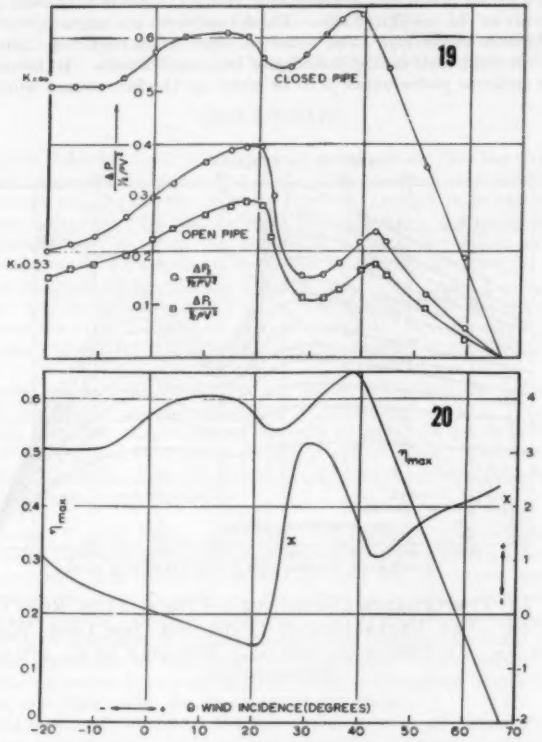


FIG. 19 (TOP)—EXPERIMENTAL RESULTS OBTAINED FOR A TYPICAL 12-IN. ROOF VENTILATOR
FIG. 20 (BOTTOM)—CHARACTERISTIC PARAMETERS x AND η_{max} FOR A TYPICAL 12-IN. ROOF VENTILATOR

2. For windspeeds above about 2 to 3 fps the relationship between the static pressure in the suction pipe and the wind velocity head is linear. This reduces the labor involved in the test procedure, since all parameters having the form $\Delta P/\frac{1}{2}\rho_0 V^2$ may be determined at only one value of the windspeed.

3. Separate determination of the ventilator resistance becomes necessary only when the stack and forced draft effects are to be evaluated.

4. Experimental determination of the magnitude of the forced draft effect is rendered impractical by its dependence on irreproducible factors such as the shape and location of the building, and the number and size of openings. This effect may, however, be evaluated theoretically if the static pressure within the building in the vicinity of the ventilator stack inlet is known.

5. Since ventilator performance is sensitive to changes in the local windspeed and direction, an accurate prediction of the extraction can be made only from a knowledge of the conditions at the ventilator site. These conditions are not necessarily the same as those in the free windstream[1], and sufficient information regarding them can be obtained only from field tests on the building or from model tests. If, however, an estimate of the ventilator performance is to be based on the free stream wind velocity, it

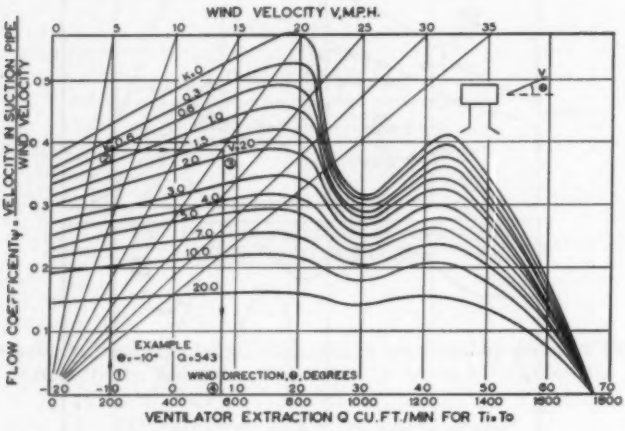

WIND VELOCITY V,M.P.H.
FLOW COEFFICIENT, VELOCITY IN SUCTION PIPE / WIND VELOCITY
EXAMPLE
WIND DIRECTION, θ, DEGREES
VENTILATOR EXTRACTION Q CU.FT./MIN FOR Ti=To

FIG. 21—PERFORMANCE CURVES FOR A TYPICAL 12-IN. ROOF VENTILATOR. THE EXTRACTION IS GIVEN FOR THE CASE WHERE $T_i = T_0$. TO OBTAIN Q_w AND Q_{wst}, MULTIPLY BY C_t AND C_tC_s RESPECTIVELY

is suggested that this velocity be employed in conjunction with the average value of the flow coefficient, the average being taken over a range of wind directions.

6. This average flow coefficient, which gives an estimate of the average magnitude of the aspiration effect, may be used as a convenient basis for comparative rating of ventilators.

7. It is estimated that the experimental procedure outlined would yield results accurate to within about 5 percent.

ACKNOWLEDGMENT

The author is indebted to the South African Council for Scientific and Industrial Research for permission to publish this paper.

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2. Experimental Studies in Natural Ventilation, by J. B. Dick (*Journal Inst. Heating and Ventilating Engrs.*, Vol. 17, No. 173, Dec. 1949).
3. Untersuchhungen von Dachluftern, by O. Schrenk (*Ergebnisse der Aerodynamischen Versuchsanstalt Gottingen*, 4 Lieferung, Munchen-Berlin 1932).
4. Die Natürliche Raumentlüftung Mittels Einzel und Flachenentlüfter, by Otto Conrad (*Zeitschrift des Verein Deutscher Ing.*, Vol. 95, No. 17/18, June 1953).

DISCUSSION

R. L. STONE†, Belmont, Calif. (WRITTEN): It seems to me that the author has presented a fundamentally sound treatment of the performance of roof ventilators and related fittings such as gas vent and chimney tops. It appears to be the most comprehensive approach developed for the functioning of these units and could certainly form a basis for development of sound test standards for both ventilators and vent tops. We plan to make use of the treatment developed by Mr. Wannenburg for study of gas vent tops in a wind tunnel recently constructed.

There is an additional point in the paper which might well be further analysed. This is the question as to whether the author's treatment of super-imposing the stack effect on the aspiration effect could be expanded to include vent gas cooling as shown by equations in use.

Another point not entirely clear from the paper is whether the constant small x introduced in Equation 10 is a true parameter for vent tops.

It would also seem desirable to expand the work to report on the performance curves of other ventilators giving their exact details of construction. The specifications of the top on which curves and Figs. 19, 20 and 21 are based should also be published either as an expanded paper or subsequently.

E. W. CONOVER, Detroit, Mich., (WRITTEN): The author is to be commended for his careful analysis of the problem and his consideration of the conditions under which roof ventilators function. It is refreshing to find someone who recognizes that there is a difference between the rated capacity and the performance when placed on a building.

About 35 years ago my Company sponsored a rather extensive program at the University of Michigan to determine the characteristics of wind and temperature difference on the natural ventilation of buildings. Several papers were presented at national meetings of this Society and the fundamentals of this investigation have been published in THE GUIDE ever since under the title Natural Ventilation.

It was found that roof ventilators functioned the same as any other opening of the same size at the same locality. With certain fenestration arrangements inflow through roof ventilators was observed when the demand for air at certain suction areas was greater than the supply through openings in pressure areas.

The tests cited in this paper appear to have been conducted on a stationary type. It is realized that this paper is on a method for testing and rating but we would like to ask the author if other than this one type has been tested, such as the pivoted or rotating types. We know of one stationary ventilator that is claimed to have positive ventilation regardless of wind direction. No doubt the curves would differ from those of Fig. 21. Would the author care to discuss this.

AUTHORS' CLOSURE: The author wishes to thank Messrs. Stone and Conover for their comments, and will attempt to discuss briefly some of the points mentioned.

---

† William Wallace Company.

Mr. Stone inquires whether the treatment of superimposing stack effect on aspiration effect could be expanded to include vent gas cooling. In answer to this question it has to be noted that, since flow in either directions through a ventilator can only be caused by a differential pressure acting across the ventilator, the first requirement for taking into account any effect such as vent gas cooling, would be to assess the magnitude of the effect in terms of its equivalent differential head. Once this head is known, the flow may be combined with that due to the aspiration effect by adding the pressure heads. It would, however, be prudent to investigate experimentally the validity of this procedure.

As regards the parameter x, its introduction was based on the following considerations: During previous tests on ventilators it was found that the resistance factor of a ventilator varied with the wind direction, this being probably due to the corresponding variation of the flow pattern within the ventilator head. A change in ventilator resistance may be considered as equivalent to a change in the effective pressure head ΔP_s which causes the flow, and the parameter x was introduced into Equation 10 to take into account this variation in pressure head. If, however, the ventilator resistance factor at a wind incidence of 0 degrees is written as $K_v + \Delta K_v$, where K_v is the *static resistance factor* (i.e. the resistance factor when the wind velocity is zero), then it may readily be shown that x and ΔK_v are equivalent. In other words, the parameter x is merely the change in ventilator resistance factor due to the effect of the wind on the flow pattern within the ventilator head.

As regards the specifications of the top for which the performance curves are given, it is of a proprietry make, and the exact specifications can unfortunately not be published. It is, however, the intention to apply the developed test method to the systematic investigation of ventilators, and the results will be published as soon as they become available. The author would be interested to see the results of the proposed tests mentioned by Mr. Stone.

Mr. Conover notes that, under certain conditions, inflow through ventilators was observed. Similar effects were noticed by the author during model tests that were conducted in conjunction with the South African National Building Research Institute. (The results are reported in reference 1, and have since been published in the *Journal* of the Institution of Heating and Ventilating Engineers, London March 1957). These tests indicated that it is indeed fallacious to consider a ventilator apart from the structure upon which it is located. Apart from reversed flow resulting from positive pressure regions on the roof of a building, a ventilator may, under certain conditions, be situated within a *stagnant zone* or *dead air region*, and under these conditions extraction through the ventilator will be caused, not by the aspiration effect, but by the static depression within the stagnant zone.

The tests reported in the paper were carried out on the stationary type of ventilator, and it may be seen from Fig. 21 that reversed flow sets in at a wind incidence of approximately 67 deg. It would, however, be quite possible to design a stationary ventilator with positive characteristics at all incidences, but the question is whether such a design could be achieved without unduly increasing the resistance.

Systematic tests on the railway coach type of ventilators were conducted by the Institute in 1953, but since the investigations were on a contract basis, the results have not been published.

1618

No. 1618

JET VELOCITIES FROM RADIAL FLOW OUTLETS

By ALFRED KOESTEL*, CLEVELAND, OHIO

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, in cooperation with Case Institute of Technology, Cleveland, Ohio

CEILING distribution of air is accomplished by air outlets which may conveniently be divided into ceiling plaques and ceiling diffusers. Diffusers and plaques usually discharge their flow parallel to the surface (wall or ceiling) in which they are mounted. This is referred to as *flush mounted*. However, the flow might also be partially free of surface interference if mounted on say, exposed ductwork. This difference in mounting is important, as it influences the performance of the outlet. A plaque has a single deflecting surface and therefore a single air passage for discharging air.† (See Fig. 1). A diffuser is sometimes defined as a manufactured and laboratory tested device for distributing air. A diffuser having multi-deflecting type air passages arranged in parallel is shown in Fig. 2. Commercial ceiling diffusers have been developed to overcome the shortcomings of the simple plaque, but both groups belong to the large class of radial flow outlets, as the simplest representatives of which may be considered the radial nozzle and the radial slot shown in Fig. 4.

In some diffusers the air discharge flow rate is not uniform in all radial directions. In this paper we are concerned with the jet velocities in a uniformly distributed circumferential air jet formed by diffusers, plaques, and radial nozzles or slots.

The basic air jet flow pattern which was analytically investigated is shown in Fig. 3. Many geometrically different types of radial flow outlets were experimentally investigated in order to determine the effect of outlet geometry on the air jet velocities and flow pattern. These outlets are listed in Table 1. The test

* Associate Professor of Mechanical Engineering, Case Institute of Technology. Member of ASHAE.

† A plaque is sometimes considered as an air distribution device manufactured in the field and similar to the one shown in Fig. 1.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Murray Bay, Canada, June 1957.

results on some of the outlets of Table 1 are also presented in Reference 1, which should also be consulted for some relevant remarks on the performance of radial flow outlets.

BASIC EQUATION

When the principle of conservation of momentum is applied to the air jet in Fig. 3, the centerline velocity (maximum jet velocity) can be found by Equation 1, which is a basic equation developed for describing the performance of all outlets belonging to the *radial flow outlet* class.

$$V_c/V_o = \frac{\sqrt{K(H_o/R_o)\cos\theta[K(H_o/R_o)\cos\theta+1]}}{\frac{\sqrt{R(R-R_o)}}{R_o}} \qquad \text{----------(1)}$$

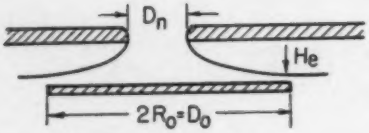

FIG. 1—PLAQUE

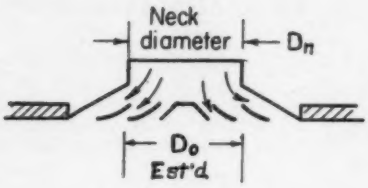

FIG. 2—DIFFUSER

Fig. 3 shows the basic flow pattern of an isothermal jet discharged from a simple radial flow outlet into a free space which is *in no way* restricted.

Since many radial flow outlets are flush mounted to a surface, the radial jet produced will flow along the surface and entrainment will take place on only one side of the jet. This increases the jet velocities by a factor of $\sqrt{2}$. See Reference 1 for a discussion of this. To find the maximum jet velocity for *flush* mounted outlets, the values found with help of Equations 1 and 2 must be multiplied by the factor $\sqrt{2}$.

The detailed mathematical analyses (on which Equation 1 is based) appear in the Appendix.

The value of the numerator of the right-hand side of Equation 1 depends on the geometric configuration of the outlet. The denominator represents the dimensionless distance from the outlet in terms of outlet radius to the point where the maximum jet velocity is V_o. For a given diffuser or a plaque Equation 1 becomes:

$$V_c/V_o = \frac{C}{\frac{\sqrt{R(R-R_o)}}{R_o}} \qquad \text{----------(2)}$$

(also see Equation A-12, Appendix) when C, a constant, can be experimentally determined, by test. V_o is computed from the total pressure drop across the diffuser if no losses are assumed. Empirical formulas describing the jet velocity behavior from

ceiling diffusers have been in use for many years. These formulas usually take the following form:

$$\text{Throw} = (\text{constant}) \times (\text{volumetric flow rate})/\sqrt{\text{Effective Area of the outlet}} \quad . \quad (3)$$

where,

Throw = distance from outlet to where the jet velocity has a predetermined value, such as, 50 fpm or 100 fpm. The constant is determined by test.

It can be shown that Equation 3 is similar to Equation 2 if the distance from the outlet is large enough so that the quantity $(R - R_o)$ is approximately equal to R.

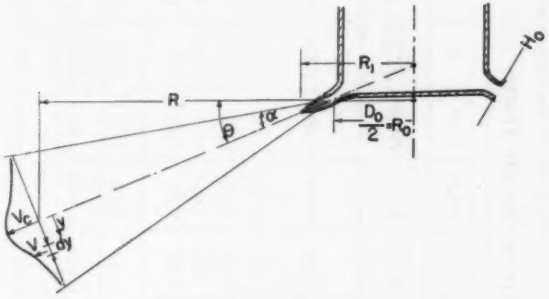

FIG. 3—BASIC RADIAL JET

The term H_o/R_o in Equation 1—where H_o is the width of the slot of the radial flow outlet shown in Fig. 3—can be easily determined for simple radial flow outlets, such as shown in Fig. 4. It is difficult to evaluate for outlets with more complex geometrical configurations, particularly commercial diffusers, but the following approach can be used for plaques. For example, consider a circular plaque outlet in which the deflecting plate is relatively far from the *neck* or nozzle as shown in Fig. 1. If it is assumed that the flow rate leaving the nozzle is equal to the flow rate issuing from the circumferential edge of the plate, (amount of entrainment in this region considered negligible), then,

$$(\pi D_n^2/4)\, V_o = V'_o\, \pi\, D_o\, H_e$$

and assuming that $V_o \approx V'_o$ (constant total head at the same ambient static pressure), this equation changes to:

$$\tfrac{1}{2}\,(D_n/D_o)^2 = H_e/R_o \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

where,

H_e/R_o = equivalant to H_o/R_o for the simple radial flow outlets shown in Fig 4.

H_e = the effective width of the jet discharged from the outlet. (See Fig. 1.)

TABLE 1—RESULTS OF OUTLETS TESTED

IDENTIFICATION No.		1	2	3	4	5	6	7	8	9	10	11	12	13
Type of Outlet (see Figs. 1, 2, 4)		Plaque												
Value of C in *Equation* 2 from jet velocity *vs.* radial distance data, Fig. 6		0.285	0.275	0.210	0.220	0.280	0.290	0.680	0.670	0.530	0.375	0.320	0.330	1.50
Dimensions of outlet in inches[a] (see Figs. 1, 2, 4)		(24) x (5 7/16) x (1 1/2)	(24) x (5 7/16) x (1)	(24) x (5 7/16) x (1 1/2)	(24) x (5 7/16) x (2)	(24) x (5 7/16) x (3)	(24) x (5 7/16) x (4)	(24) x (4 x 4) x (1 1/2)	(24) x (4 x 4) x (1)	(24) x (4 x 4) x (1 1/2)	(24) x (4 x 4) x (2)	(24) x (4 x 4) x (3)	(24) x (4 x 4) x (4)	(5 7/16) x (5 7/16) x (1 1/2)
H_0/R_0		0.0416	0.0832	0.125	0.167	0.250	0.333	0.0417	0.0832	0.125	0.167	0.025	0.333	0.184
H_e/R_0	By *Equa.* 4	0.0258	0.0258	0.0258	0.0258	0.0258	0.0258	0.0178	0.0178	0.0178	0.0178	0.0178	0.0178	0.500
	By *Equa.* 5													

IDENTIFICATION No.		14	15	16	17	18	19	20	21	22	23	24	25	26
Type of Outlet (see Figs. 1, 2, 4)		Plaque							Radial Slot 3[b]	Radial Nozzle 3[b]				Diffuser
Value of C in *Equation* 2 from jet velocity *vs.* radial distance data. Fig. 6		1.27	1.27	0.590	0.550	0.600	1.55	1.80	1.00	1.05	0.400	0.340	0.490	0.85
Dimensions of outlet in inches[a] (see Figs. 1, 2, 4)		(5 7/16) x (5 7/16) x (1)	(5 7/16) x (5 7/16) x (1 1/2)	(14) x (5 7/16) x (1/2)	(14) x (5 7/16) x (1)	(14) x (5 7/16) x (2)	(14) x (14) x (1)	(14) x (14) x (2)	(14) x (1)	2.32 x (1/4)	(11 1/2) x (1/2), Nozzle angle 45°	(11 1/2) x (1/4), Nozzle angle 45°	(11 1/2) x 0.219; Nozzle angle 45°	8-inch neck diam. $\theta = 25°$ (see Fig. 2)
H_0/R_0		0.369	0.553	0.0714	0.143	0.286	0.143	0.286	$C_d = 0.6$ 0.0857	$C_d = 1.0$ 0.216	$C_d = 1.0$ 0.087	$C_d = 1.0$ 0.0435	$C_d = 1.0$ 0.0381	— —
H_e/R_0	By *Equa.* 4	0.500	0.500	0.0755	0.0755	0.0755	0.500	0.500	—	—	—	—	—	0.5
	By *Equa.* 5													N = 3 0.167

[a] For plaques: Plaque diameter in inches; Nozzle diameter in inches; Distance between plaque and face of nozzle in inches.
[b] For radial slots or nozzles: Outer duct or pipe diameter in inches; Width of slot or nozzle in inches.

If the deflecting plate of the plaque type outlet of Fig. 1 is relatively near the neck of the nozzle so that the width of the jet is equal to this distance, H_o, Equation 4 is not applicable since V_o is not equal to V'_o when the flow is restricted in this manner. In this case the term H_o/R_o is less than the term H_e/R_o, as computed from Equation 4, and should be used in Equation 1.

Example: A flow of 500 cfm from a nozzle 1 ft in diameter impinges on a plaque located 6 in. from the face of the nozzle. The nozzle face is mounted flush in a ceiling. Find the maximum jet velocities 10 ft from the center of the plaque if the plaque diameter is (a) 2 ft in diameter, (b) 4 ft in diameter. Assume K equal to 4 and θ equal to zero in both cases. V_o through the nozzle is $500/\pi\ (\frac{1}{4})^2 = 606$ fpm in both cases which is also approximately equal to the jet velocity leaving the edge of the plaque if losses are negligible.

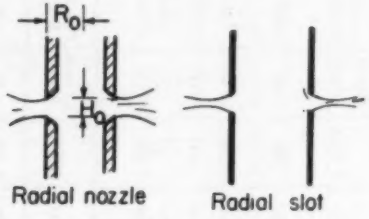

FIG. 4—RADIAL OUTLETS

Solution (a):

$H_e/R_o = \frac{1}{2}\ (\frac{1}{2})^2 = 0.125$ which is less than
$H_o/R_o = (6/12)/1 = 0.5$ which indicates that the width of the jet is less than 6 in. which makes Equation 4 applicable.

Applying Equation 1,

$$V_c/636 = \frac{\sqrt{4\times 0.125[4\times 0.125+1]}}{\frac{\sqrt{10(10-1)}}{1}}$$

$$= 0.866/9.49$$

$V_c = 58.0$ fpm if the jet were totally free, and $V_c = 58 \times \sqrt{2} = 82$ fpm since the jet flows adjacent to a surface. *Note*: C in Equation 2 is 0.866 for this case.

Solution (b):

$H_e/R_o = \frac{1}{2}\ (\frac{1}{4})^2 = 0.0312$ which is again less than
$H_o/R_o = (6/12)2 = 0.25$ making Equation 4 applicable.

Applying Equation 1,

$$V_c/636 = \frac{\sqrt{4\times 0.0312\,[4\times 0.0312+1]}}{\frac{\sqrt{10(10-2)}}{2}}$$

$$= 0.375/4.47$$

V_c = 53.4 fpm if the jet were totally free, and $V_c = 53.4\sqrt{2} = 75.5$ fpm since the jet flows adjacent to a surface. *Note*: C in Equation 2 is 0.375 for this case.

The results show that the maximum jet velocities 10 ft from the plaque are practically the same in both cases. This is understandable since the initial momenta are the same. It is the differences in H_c/R_o and R_o which contribute to the small difference in jet velocities for the 2 plaques.

PLAQUE WITH MULTI-DEFLECTING PLATES

Now consider a plaque similar to Fig. 5 with N plates and apply the continuity equation, and

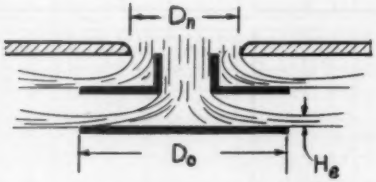

FIG. 5—PLAQUE WITH DEFLECTING PLATES IN PARALLEL

$$\pi\ D_n^2/4 = \pi\ D_o\ N\ H_c$$

providing the plates are equal in diameter, the distances H_c equal in magnitude, and the velocities of the jets leaving the edge of each plate are the same. This simplification results in:

$$\tfrac{1}{2}\ (D_n/D_o)^2\ (1/N) = H_c/R_o \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

where

H_c/R_o = equivalent to H_o/R_o with the same restrictions as previously pointed out.

If each deflecting plate in Fig. 5 has a different diameter as is the case for typical diffusers, then R_o or D_o in Equation 5 might represent some mean value.

It should be realized that actual diffusers used in practice produce flow patterns on their deflecting plates which are more complex than the idealized situation illustrated in Fig. 5.

Theoretically adding deflecting plates in parallel as in Fig. 5, reduces H_c/R_o for a fixed flow rate, and according to Equation 1, this reduces the ratio, V_c/V_o at a given distance from the outlet. This reasoning applies if the individual radial jets produced per deflecting plate do not coalesce into a single jet. Actually the jets will merge at a certain distance from the outlet (see Reference 2) and as a result the effect produced by adding deflecting plates or cones will be minimized.

RESULTS

In Fig. 6 the test results obtained for each outlet described in Table 1 are plotted in accordance with Equation 2. The experimentally determined value of C of

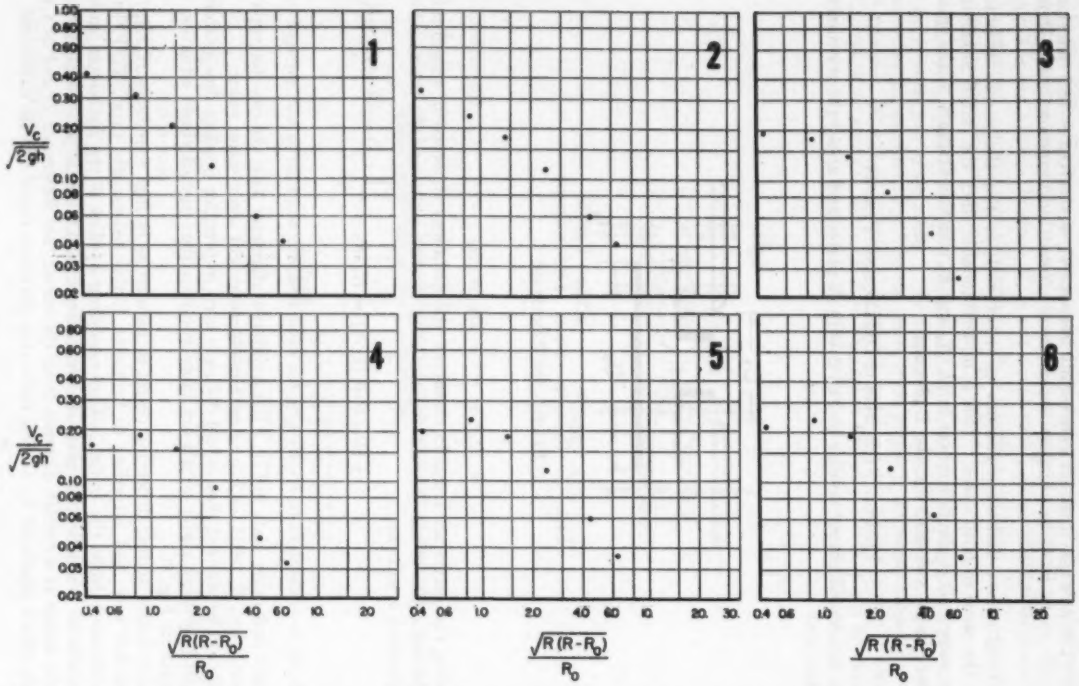

FIG. 6—MAXIMUM VELOCITIES VERSUS RADIAL DISTANCE FROM OUTLET. OUTLETS ARE IDENTIFIED IN TABLE 1

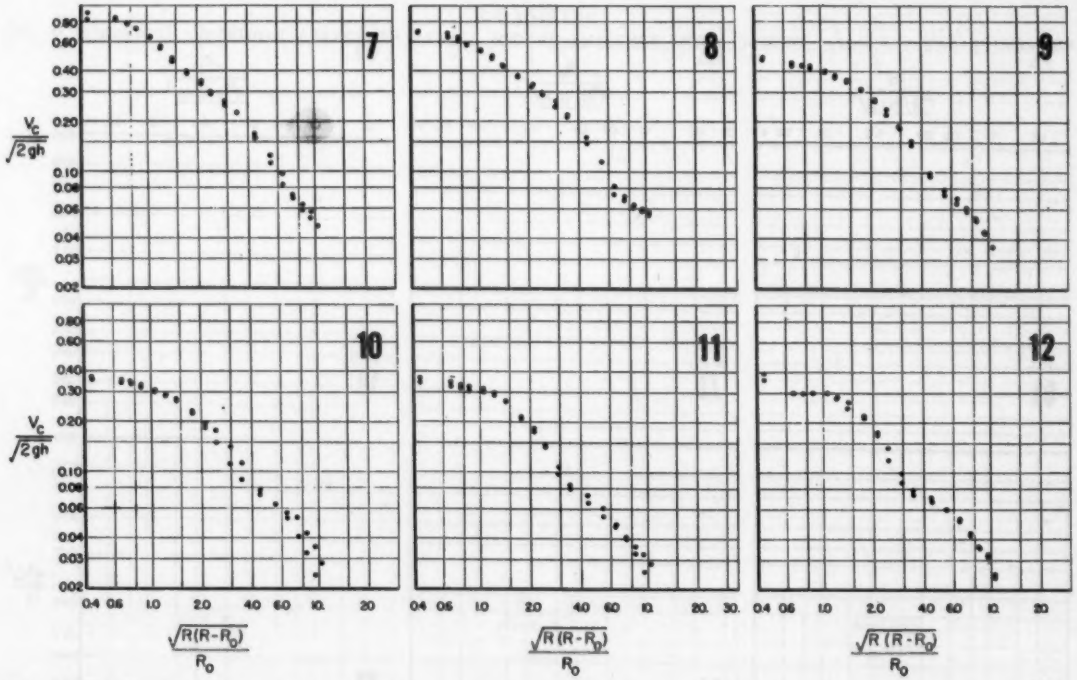

FIG. 6 (*continued*)—MAXIMUM VELOCITIES VERSUS RADIAL DISTANCE FROM OUTLET. OUTLETS ARE IDENTIFED IN TABLE 1

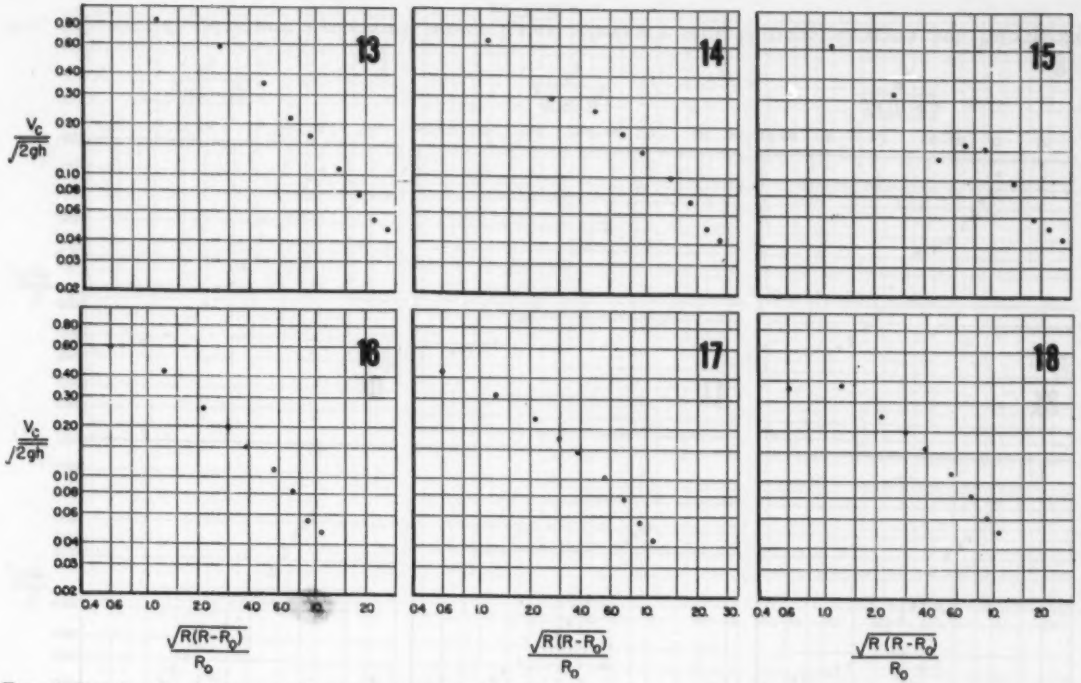

FIG. 6 (*continued*)—MAXIMUM VELOCITIES VERSUS RADIAL DISTANCE FROM OUTLET. OUTLETS ARE IDENTIFIED IN TABLE 1

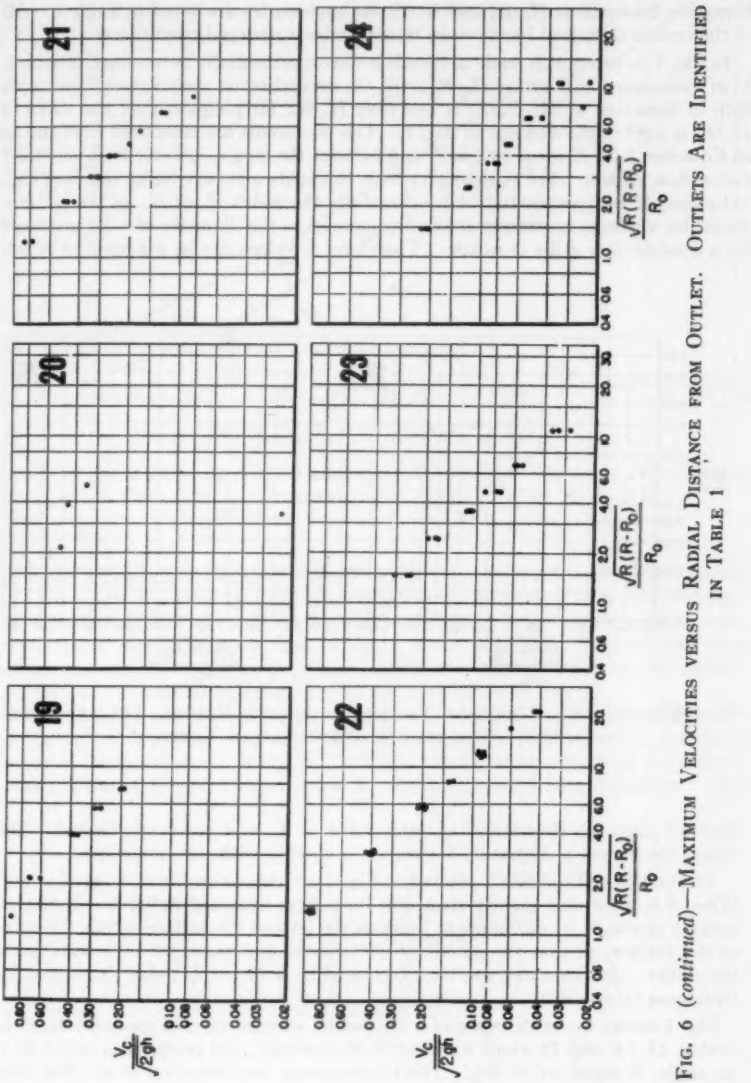

FIG. 6 (*continued*)—MAXIMUM VELOCITIES VERSUS RADIAL DISTANCE FROM OUTLET. OUTLETS ARE IDENTIFIED IN TABLE 1

Equation 2 as well as H_o/R_o and H_e/R_o for each outlet are listed in Table 1. All of the outlets described herein were tested under isothermal conditions.

In Fig. 7 an attempt is made to correlate the experimentally determined constant, C, of Equation 2 with either H_o/R_o or H_e/R_o depending on their values found with help of Equation 4. If H_e/R_o is less than H_o/R_o, for plaques, then the value of H_e/R_o is used in the abscissa of Fig. 7. The test points are compared with curves of Equation 1 for K equal to 4 with and without the factor, $\sqrt{2}$, for flush mounted radial flow outlets. For outlets with high Reynolds numbers (large size and high velocities) K is approximately 6.5. Normally the width H_o or H_e, of the jet produced by a plaque or diffuser is small compared to the diameter of a jet produced by a straight flow grille or nozzle. Therefore, K values can be expected to be less

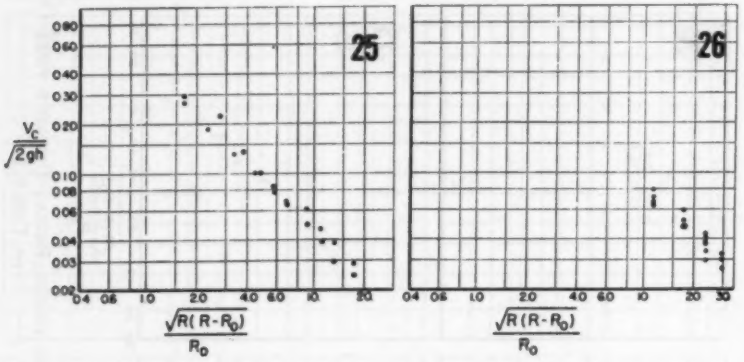

FIG. 6 (*concluded*)—MAXIMUM VELOCITIES VERSUS RADIAL DISTANCE FROM OUTLET. OUTLETS ARE IDENTIFIED IN TABLE 1

than 6.5 (also see Reference 1), and a value of $K = 4$ has been selected. This makes the curves of Equation 1 compare favorably with the test points.

The curves of Equation 1 plotted in Fig. 7 are determined with θ equal to zero When θ is somewhat greater than zero for a flush mounted radial flow outlet the suction produced by entrainment between the jet and the surface, draws the jet up to the surface, so that the jet effectively behaves as if there were no deflection at the outlet. Jet velocities are then increased by a factor $\sqrt{2}$ due to reduced entrainment on one side.

Fig. 8 shows the coalescence of a jet due to vacuum in void space produced by outlets 23, 24, and 25 which were not flush mounted, and produced a radial jet at an angle, θ, equal to 45 deg. This phenomenon was observed at all flow rates which indicates that this phenomenon depends on the geometry of the outlet. When the outlet was flush mounted and a small hole drilled in the deflecting plate so that supply air can fill the void without creating a suction, the radial jet was drawn up to the mounting surface due to the suction produced in the region between

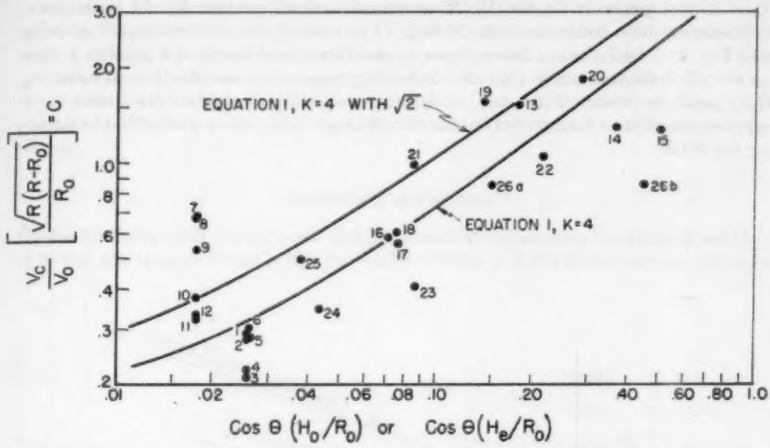

FIG. 7—CORRELATION OF TEST DATA FROM 26 OUTLET TYPES IN ACCORDANCE WITH EQUATION 1. NUMBERS WITH POINTS REFER TO TABLE 1

the radial jet and the surface. Although outlets Nos. 23, 24, 25, have radial nozzles set at an angle of 45 deg (see Fig. 8), the free jet at a distance from the discharge face flowed along the wall as if emanating from a radial outlet with θ equal to zero. Therefore, in plotting the test data in Fig. 7, $\theta = 0$ was used. Figs. 6 and 7 also contain test results for the only commercial diffuser tested (outlet No. 26), which is a typical commercial outlet similar to the one shown in Fig. 2.

Outlet No. 26 was the only outlet tested which formed a radial jet with an effective deflection greater than zero. The outlet was mounted exposed in a manner similar to the outlet shown in Fig. 8. In Fig. 7 two points are shown for the plotted symbol of outlet No. 26. Point 26*a* is for H_e/R_o computed from Equation 5 with

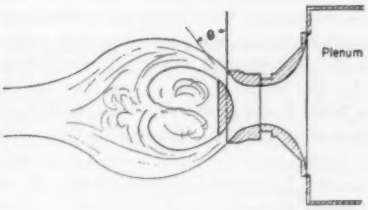

FIG. 8—EXPOSED RADIAL NOZZLE (NOT FLUSH MOUNTED IN WALL) SHOWING COALESCENCE OF JET DUE TO VACUUM IN VOID SPACE

$N = 3$, and point 26*b* is for H_e/R_o computed from Equation 4. In both cases, θ, (measured flow deflection) was 25 deg. The ratio D_n/D_o was estimated as unity (see Fig. 2). Since point 26*a* is closer to the theoretical curve of Equation 1 than point 26*b*, indications are that the deflecting cones may be effective in reducing H_e/R_o and the *throw*. However, no definite conclusion should be drawn since some experimenters have found that multiple deflecting cones are not effective in reducing the *throw*.

COMMERCIAL DIFFUSERS

Manufacturers of commercial diffusers publish test data on their products which as a rule are not presented in a manner which permits a direct comparison between

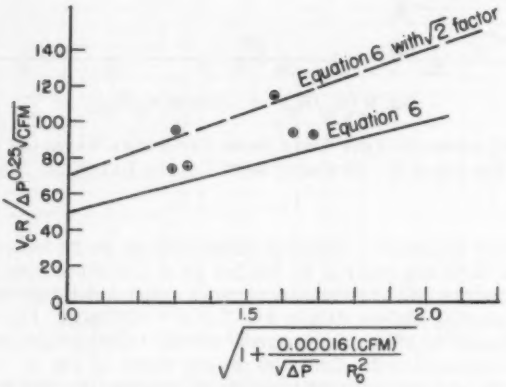

FIG. 9—COMPARISON OF EQUATION 6 WITH DATA OBTAINED FROM A DIFFUSER MANUFACTURER

these data and the equations presented in this paper. However, by judicious manipulation of Equation 1, a comparison can be made with catalog data.

Normally manufacturers present their *throw* data (distance from outlet to a prescribed terminal velocity) as a function of flow rate in cubic feet per minute and air pressure behind outlet in inches of water. If the following substitutions are made into Equation 1:—$A_e = 2\pi R_o H_e$; $A_e = \text{cfm}/V_o$; $V_o = 4005\sqrt{\Delta P}$ (for standard air density of 0.075 lb per cu ft); $K = 4.0$; $\sqrt{R(R - R_o)}$ is approximately equal to R if R_o is small contrasted to R; and Cos $\theta = 1$, we obtain Equation 6, thus

$$\frac{V_c R}{\Delta P^{0.25}\sqrt{cfm}} = 50.3\sqrt{1+\frac{0.00016\times(cfm)}{\sqrt{\Delta P}\ R_o^2}} \quad \text{----(6)}$$

Equation 6 is plotted in Fig. 9 with and without the $\sqrt{2}$ multiplying factor applied to the right side of Equation 6. Published data obtained from a manufacturer's catalog are also plotted (symbols) for comparison with Equation 6. The plotted data represents: (1) 3 geometrically different diffuser types; (2) 240 to 7600, (cfm); (3) 0.122 to 0.249 in. of water, (ΔP); (4) 3 to 5 deflecting cones, (N); (5) 100 to 200 fpm, (V_o); (6) 7 to 42 ft, R; and (7) 8 to 38 in., diffuser neck diameter.

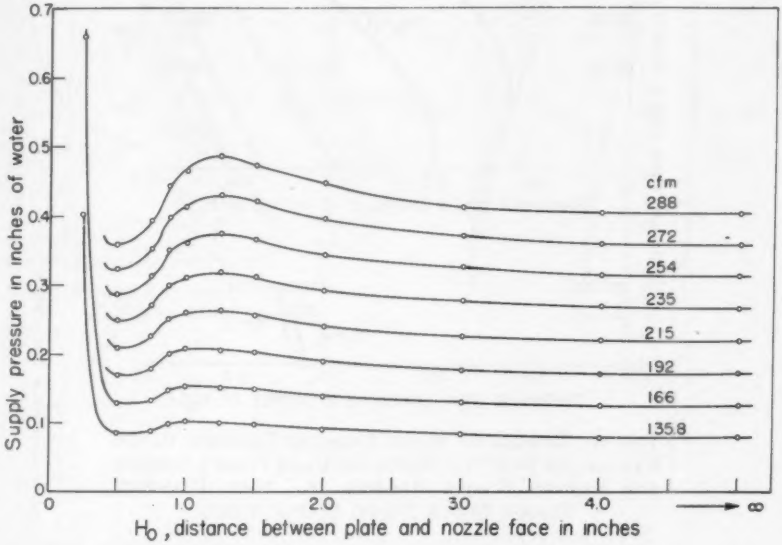

FIG. 10—EFFECT OF DISTANCE, H_o, ON SUPPLY PRESSURE FOR A 24-IN. DIAMETER PLAQUE WITH A $5\frac{7}{16}$-IN. NOZZLE

R_o in Equation 6 was arbitrarily taken as one-half the neck diameter in feet. According to the catalog the diffusers were all flush mounted in the ceiling.

The effect of N (number of deflecting cones) on the product (V_oR) in Fig. 9 or in Equation 6 was assumed minor. The data plotted in Fig. 9 indicated that this was the case and that Equation 5 was not applicable in the derivation of Equation 6. This contradicts the implication made in contrasting point 26*a* with point 26*b* in Fig. 7 for outlet No. 26 which is a commercial diffuser with 3 cones.

SOME EXPERIMENTAL OBSERVATIONS

Several calibrated anemometers were used to measure the jet velocities; *i.e.*, thermal-anemometers, bridled vane anemometers and total head tubes. As many

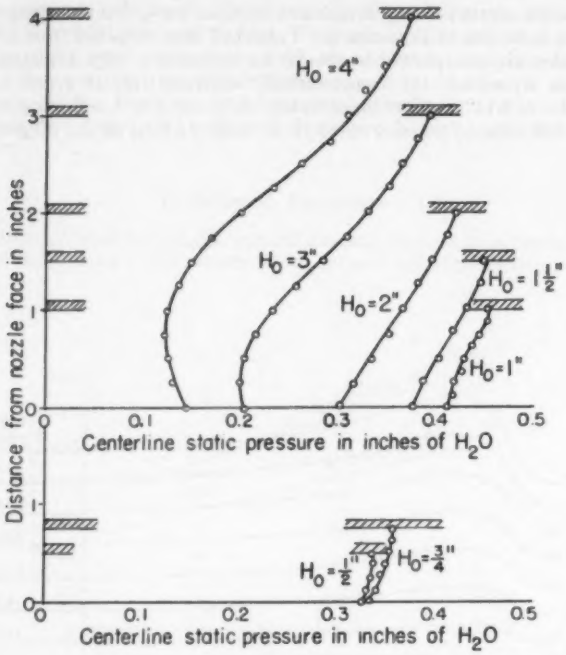
Distance from nozzle face in inches
H_0 = 4"
H_0 = 3"
H_0 = 2"
H_0 = 1 1/2"
H_0 = 1"
H_0 = 1/2"
H_0 = 3/4"
Centerline static pressure in inches of H_2O
Centerline static pressure in inches of H_2O

FIG. 11—RESULTS OF STATIC PRESSURE TRAVERSE ALONG CENTERLINE BETWEEN NOZZLE FACE AND PLAQUE SURFACE FOR VARIOUS PLAQUE HEIGHTS, H_o. 24-IN. DIAMETER PLAQUE WITH A 5 7/16-IN. SUPPLY NOZZLE

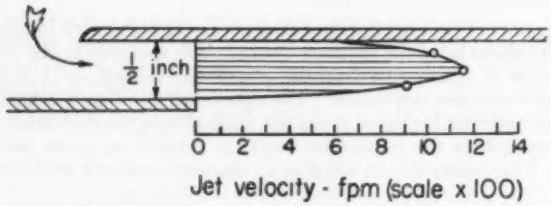

FIG. 12—JET VELOCITY PROFILE AT EDGE OF PLAQUE. 24-IN. DIAMETER PLAQUE WITH A 5 7/16-IN. DIAMETER SUPPLY NOZZLE. H_o EQUAL TO 1/2 IN.

as 6 observers were involved during a period of 3 years of testing and the instrument used depended on the preference of the individual.

During the tests many detailed measurements were made which, if presented in this paper, would be too voluminous. Therefore, only some representative measurements of interest are herein reported. In Fig. 10 is shown the effect of plaque height, H_o, on the required nozzle pressure behind the outlet for various flow rates

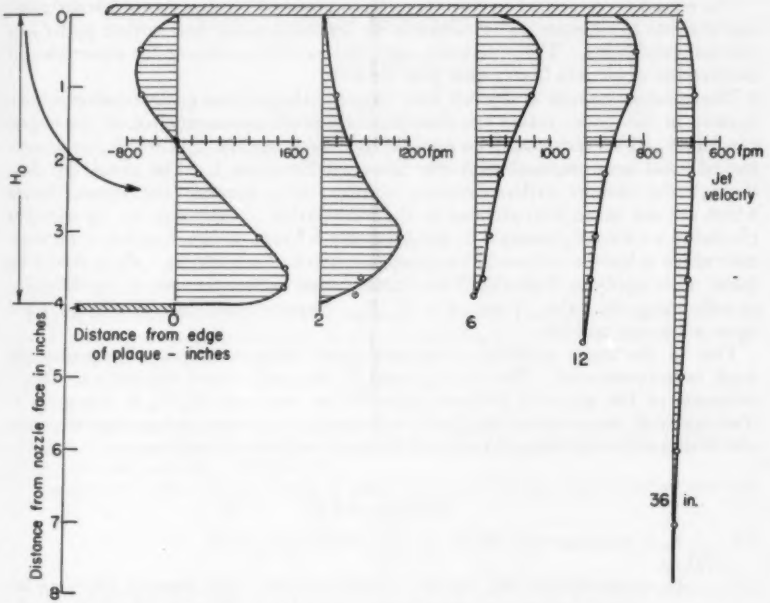

FIG. 13—VELOCITY PROFILES PRODUCED BY A JET FROM A 24-IN. DIAMETER PLAQUE WITH A 4 BY 4-IN. SQUARE NOZZLE; H_o EQUAL TO 4 IN.

impinged on a 24-in. diameter plaque. The characteristic *dip* in pressure at a plaque height of ½ in. as shown in the figure is typical of plaque type outlets. The *dip* was observed at a plaque height of ¾ in. when a 14-in. diameter plaque was placed in front of the nozzle.

In Fig. 11 are shown centerline static pressure traverses made between the face of the nozzle and the face of the plaque.

Velocity profiles were measured near the edge of the plaques. When the plaque height was small the velocity profiles were uniform as shown in Fig. 12 for a height of ½ in. with a 24 in. diameter plaque and a 5 7/16-in. diameter nozzle. As the plaque height is increased a turbulent region forms adjacent to the wall behind the plaque and extending a small distance from the edge of the plaque. This is illustrated in Fig. 13 for a plaque height of 4 in. (24-in. diameter plaque; 4-in. by 4-in. square

nozzle). The flow from the nozzle is directed along the surface of the plaque and a counterflow (negative velocities shown in Fig. 13) stream is indicated near the wall. When the flow is clear of the plaque the free jet is drawn to the wall surface due to vacuum created in the turbulent or counterflow regions. Some velocity profile data on plaque type outlets were also reported in Reference 1.

CONCLUSIONS AND RECOMMENDATIONS

The equations presented in this paper form a basis for a designer of air distribution systems to estimate air velocities in air jets from radial flow outlets under isothermal conditions. The equations may also provide guidance for experimental evaluations of air jets from radial flow outlets.

The complex nature of the air flow through the various outlet configurations covered in this paper makes the theory only a crude approximation of the actual conditions. This can readily be realized by examining Fig. 7 in which the experimental data are compared with the curves of Equation 1. The actual jet flow through the various outlets involves complex eddy, turning, and impact losses which are not taken into account in the formulation of Equation 1. In addition the radial jet theory presented in the Appendix is based on conservation of momentum which is known to be only an approximation in a free air jet. Care should be taken when applying Equation 1 to practical type outlets because of the difficulty of estimating the value of the term H_o/R_o. Experimental data should be relied upon whenever possible.

Due to the many possible configurations of diffusers additional experimental work is recommended. The work should be directed toward the accurate determination of the effect of diffuser geometry on the term H_o/R_o in Equation 1. The material presented in this paper is extensive in nature rather than intensive which should be the philosophy for future work in radial flow outlets.

NOMENCLATURE

V_o = maximum velocity in air jet, fpm feet per minute.

$\sqrt{2gh}$ or

V_o = maximum outlet velocity (computed from total pressure difference between supply air and ambient room air), i.e., $V_o = \sqrt{2gh}$ where h is the total air pressure in feet of air.

V'_o = velocity leaving plate edge of plaque-type outlet, see Fig. 1, fpm.

R = distance from geometric center of outlet to where maximum jet velocity is V_o, feet.

R_o = radius of outlet, feet.

H_o = width of slot opening in outlet, or distance between plate surface and nozzle face for plaques, feet.

D_o = diameter of plaque, feet or inches.

D_n = diameter or equivalent diameter of supply opening behind plaque, feet.

H_e = effective width of jet being discharged from outlet, feet.

N = number of conical type deflectors in parallel in a diffuser on which air is radially deflected.

θ = horizontal deflection of air stream, degrees.

C_d = coefficient of discharge, dimensionless.

C = constant.

K = length of the constant velocity core in terms of the width, H_o, of the circular slot (or in terms of the diameter for a jet issuing from a simple nozzle). If the Reynolds number of the jet evaluated at the outlet is large enough, $K = 6.5$ diameters for a nozzle, and $K = 6.5$ slot widths for a slot, dimensionless. See Reference 1.

ΔP = pressure drop across outlet in inches of water.
cfm = air-flow rate in cubic feet per minute.

ACKNOWLEDGMENTS

The author is indebted to Prof. G. L. Tuve for his help and suggestions. The experimental work was performed over a period of several years by K. Kjolner, E. Jones, W. Kerka, E. Cox, and B. Davis, students in mechanical engineering. The figures were prepared by Mrs. Ruth Strahosky.

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APPENDIX

MOMENTUM ANALYSIS

It has been demonstrated that the Gauss error-function or normal probability type curve is an accurate representation of the cross-sectional velocity profile of a free jet. Therefore, one can let an equation of the form,

$$V = V_c e^{-a y^2} \quad \text{(A-1)}$$

define the shape of the velocity profile of the free radial jet shown in Fig. 3. The factor a can be called a shape factor since its value determines the width of the profile for a given value of centerline velocity, V_c.

The stream momentum, M, at any arbitrary distance from an apparent point source is,

$$M = 2\pi R \rho \int_{y=-\infty}^{y=+\infty} V^2 \, dy \quad \text{(A-2)}$$

where ρ is assumed constant throughout the field of flow. The apparent point source is assumed to lie on the face of nozzle as shown in Fig. 3.

If the function of V (Equation A-1) is substituted into Equation A-2, then

$$M = 2\pi R \rho V_c^2 \int_{y=-\infty}^{y=+\infty} e^{-2a y^2} \, dy. \quad \text{(A-3)}$$

If the limits of y are changed to $y = 0$ to $y = \infty$, the jet momentum, M, is multiplied by one-half i.e.,

$$\tfrac{1}{2}M = 2\pi R \rho V_c^2 \int_{y=0}^{y=+\infty} e^{-2a y^2} \, dy \quad \text{(A-4)}$$

Integrating Equation A-4 results in the following equation:

$$M = 2\pi R \rho V_c^2 \sqrt{\frac{\pi}{2a}} \quad \text{(A-5)}$$

In order to determine a as a function of distance from the radial nozzle, a constant angle of expansion, α, for the velocity profile is assumed such that V/V_c in Equation

A-1 has a constant value at

$$y = (R - R_0)\frac{\tan \alpha}{\cos \theta}.$$

Substituting this expression for y into Equation A-1 and differentiating,

$$d\left(\frac{V}{V_e}\right) = 0 = d(e^{-a(\tan\alpha/\cos\theta)^2(R-R_0)^2})$$

$$0 = e^{-a(\tan\alpha/\cos\theta)^2(R-R_0)^2}\left[-a\left(\frac{\tan\alpha}{\cos\theta}\right)^2[2RdR - 2R_0\,dR] - \left(\frac{\tan\alpha}{\cos\theta}\right)^2(R - R_0)^2\,da\right]$$

simplifying

$$0 = \frac{2\,dR}{R - R_0} + \frac{da}{a}$$

integrating

$$0 = 2\ln(R - R_0) + \ln a + C_1$$

or

$$a = \frac{C_2}{(R - R_0)^2} \quad \ldots\ldots\ldots\ldots \quad \text{(A-6)}$$

Substituting Equation A-6 into Equation A-5,

$$M = 2\pi R\,\rho\,V_e^2(R - R_0)\sqrt{\frac{\pi}{2C_2}} \quad \ldots\ldots \quad \text{(A-7)}$$

If the jet momentum is conserved, then the momentum at R equal to R_1 (radial distance to termination of constant velocity core where V_e equals V_o, see Fig. 3) can be equated to the right hand side of Equation A-7 as follows:

$$2\pi R_1\,\rho\,V_0^2(R_1 - R_0)\sqrt{\frac{\pi}{2C_2}} = 2\pi R\,\rho\,V_e^2(R - R_0)\sqrt{\frac{\pi}{2C_2}}$$

and

$$R_1\,V_0^2(R_1 - R_0) = R\,V_e^2(R - R_0) \quad \ldots\ldots \quad \text{(A-8)}$$

For a slot type outlet such as is under consideration here, the length of the constant velocity core can be expressed in terms of slot widths, H_o; i.e.,

$$\frac{R_1 - R_0}{\cos\theta} = K\,H_0 \quad \ldots\ldots\ldots\ldots \quad \text{(A-9)}$$

where K is the core length in terms of H_o. There are experimental indications that K is a function of the Reynolds number evaluated at the discharge nozzle only when the Reynolds number is low. For a Reynolds number greater than about 3×10^4, values of K will be greater than 6.0. Values of K can be expected to lie between 6.5 and 4.0 where the lower values are for small size outlets and low velocities. Normally, a radial flow outlet can be classed as small in size since the characteristic dimension, H_o, will be small compared to the equivalent diameter of most grilles. Therefore, the value of K for radial flow outlets can usually be expected to be less than the value of K for straight flow outlets when both outlets have equal flow rates and discharge velocities.

Substituting Equation A-9 into Equation A-8,

$$V_e/V_0 = \frac{\sqrt{K(H_0/R_0)\cos\theta[(K\,H_0/R_0)\cos\theta + 1]}}{\frac{\sqrt{R(R - R_0)}}{R_0}} \quad \ldots \quad \text{(A-10)}$$

For a given type of radial flow outlet Equation A-10 states that,

$$V_c/V_0 \frac{\sqrt{R(R - R_0)}}{R_0} = \text{constant} \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(A-11)}$$

and at far distances from the outlet where $(R - R_0)$ is approximately equal to R Equation A-11 becomes,

$$V_c/V_0 = \frac{\text{constant } R_0}{R} = \frac{C_2}{R} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(A-12)}$$

where C_2 is equal to R_0 times the constant in Equation A-11.

APPENDIX NOMENCLATURE

V = Velocity at y, feet per second.
V_c = Maximum centerline velocity at $R/\cos\theta$, feet per second.
V_0 = Outlet velocity, feet per second.
y = Distance from centerline, feet.
R = Horizontal radial distance, feet.
R_1 = Horizontal radial distance to termination of constant velocity core, feet.
R_0 = Horizontal radius of outlet, feet.
a = Shape factor for velocity profile.
M = Jet momentum, pounds.
C_1, C_2, C_3 = Constants.
K = Core length in terms of H_0, dimensionless.
H_0 = Width of radial nozzle, feet.
θ = Angle of jet centerline measured from the horizontal, radians.
α = Angle of expansion of jet, radians.
ρ = Density, slugs per cubic foot.

DISCUSSION

D. J. INGALLS AND S. F. GILMAN, Syracuse, N. Y. (WRITTEN): The author is to be commended on his extensive treatment of radial flow outlets. It would seem from the graphs of Fig. 6 that Equation 2, $\frac{V_c}{V_0}\frac{\sqrt{R(R - R_0)}}{R_0} = C$, predicts the velocity pattern very closely if C, which includes both the effects of K and $\frac{H_0}{R_0}$, is experimentally determined and known. If C could be determined for various radial flow geometries then there would be little error in the prediction of velocities by the designer except at small distances from the outlet. Does the author feel it would be helpful if the diffuser manufacturers obtained values of C for their products?

Perhaps because of the difficulty of selecting a value of R_0 for commercial diffusers, more research such as this is needed.

The value of $C = 0.575$ given in Table 1 for plaque 10 should probably be 0.375.

Assuming the latter value, one may note how C varies as a function of H_0 for plaques 1 through 6 and 7 through 12. By comparison of the experimental values, C decreases to a minimum and then increases in both cases. Whether this is accidental or has some basis is not known. Perhaps Professor Koestel would care to theorize on this point.

W. O. HUEBNER, New York, N. Y. (WRITTEN): Professor Koestel's paper is another welcome addition to our knowledge on the performance of air distribution devices, and it again indicates the difficulties encountered in attempting to merge the gap between the theory of air distribution and the necessities of practical application needs.

Professor Koestel has clearly defined the limitations of his formula:

1. It deals only with *isothermal* conditions and with straight *free* jets—whereas in practice we usually have a difference between the supply air temperature and the room

air temperature and enclosing walls and other restrictions. This changes the air pattern considerably.

2. Also, the theory itself rests on a multiplicity of simplifying assumptions. These are necessary in order to arrive at equations which not only can be written but also easily be solved.

Manufactured radial flow devices—ordinary diffusers—are so arranged that they break up the mass of supply air and influence its discharge with the aim of creating in the conditioned room a steady air motion—preventing stagnation of air on the one hand and drafts on the other. Therefore, the geometric configuration of the air distribution device is highly important, but the determination of its constant may not always be easy.

Professor Koestel has presented a rather complete series on the behavior of the various air distribution methods. This paper—a pioneering one—is a most welcome contribution, and we are grateful to Professor Koestel for putting the performance of the cloud of radial flow outlets on a rational basis.

I wholeheartedly agree with Professor Koestel on the need of further experimental work in the field. The investigation of the influence of the geometric configuration of specific outlets on their performance, however, it seems to me should be left to the manufacturer rather than done by a scientific institution. They now have a fine basis to start from.

AUTHOR'S CLOSURE: I am grateful to D. J. Ingalls and S. F. Gilman for their comments. The value of C for plaque 10 in Table 1 should be 0.375 instead of 0.575.

Yes, it would be helpful if the diffuser manufacturer would obtain the value for C for their products; however, other factors must then be taken into consideration such as: (1) effect of enclosure on jet performance (2) gravity effects in non-isothermal jets (3) variation of C in the low Reynolds number range.

More research is needed on this subject especially in determining the relationship between the geometry of the outlets and C.

In Table 1, C is observed to vary with H_0 for plaques 1 through 6 and 7 through 12. This is possibly due to the variation of losses (such as turning and eddy) with outlet geometry. The theory does not explain this. More extensive experimental data would be required for a correlation.

To Mr. Huebner I offer thanks for his discussion. It will make an excellent addition to the discussion of the paper. I am inclined to agree with Mr. Huebner's viewpoint that the manufacturer should handle the question of "how do changes in outlet configuration affect performance?" However, they should also publish their findings for all to benefit.

1619

No. 1619

VELOCITIES AND TEMPERATURES ON AXIS OF DOWNWARD HEATED JET FROM 4-INCH LONG-RADIUS ASME NOZZLE

By RUDIGER KNAAK*, MANHATTAN, KAN.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, in cooperation with the Engineering Experiment Station of Kansas State College, Manhattan, Kan.

THE research reported in this paper was pursued at Kansas State College under the guidance of Linn Helander and the Technical Advisory Committee on Air Distribution.† The research was limited in scope. In the first place, it dealt solely with a downwardly projected, heated jet formed by a 4-in. long-radius *ASME* nozzle. In the second place, its specific objective was to obtain and correlate data on the centerline velocity, U_c, and the centerline temperature elevation, $T_c - T_a$.

Measured valves of U_c and $T_c - T_a$ were correlated with the travel of the jet by means of 2 curve-matching equations, one for the centerline velocities, the other for the centerline temperatures (Equations 4 and 11). The equation used for the centerline velocity, U_c, involved 4 interrelated empirical factors: (1) h_0, which located the apparent point source of the jet; (2) L_p and (3) $L_{max,\,n}$ which located the points of travel at which U_c from the equation had, respectively, the outlet value, U_{0c}, and a value of zero; and (4) an exponent, n. By a process of trial and error, combinations of these factors consistent with the experimental data were established for values of n between ⅓ and unity. Each value of n was associated with a unique value of $L_{max,\,n}$. The best correlation of the data was obtained by setting n equal to ½. In this case, $L_{max,\,n}/D_0$ had a value of $(2B_{0c})^{1/2}$. (D_0 = outlet diameter, B_{0c} = outlet buoyancy number, see Nomenclature.) When n was

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* Dr. Eng., Bremen, Germany. Visiting Instructor, Department of Mechanical Engineering, Kansas State College, during the year 1954–55.

† Personnel: W. O. Huebner, *chairman*; H. R. Limbacher, *vice chairman*; P. A. Argentieri, H. F. Brinen, J. C. Davidson, S. H. Downs, S. F. Gilman, Linn Helander, F. B. Holgate, W. W. Kennedy, W. B. Kirk, Alfred Koestel, E. J. Kurek, J. N. Livermore, R. D. Madison, L. G. Miller, H. B. Nottage, L. R. Phillips, C. H. Randolph, N. S. Shataloff, R. D. Tutt, Hrant Yousoufian.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Murray Bay, Que., Canada, June 1957.

set equal to unity, $L_{max,\,n}/D_0$ had the value of L_{max}/D_0 previously reported[2], namely, $1.66(B_{0c})^{1/2}$. Good correlation of U_c with travel was obtained with all values of n from ⅓ to unity, when n was coupled with the corresponding best correlating value of $L_{max,\,n}$.

The empirical equations based on n having a value of ½ are summarized in Table 1. $L_{max,\,n}$ in this table is not the same as L_{max}, the sensible throw of the jet. For throws in excess of 8 outlet diameters the latter is properly evaluated by:

$$(L_{max}/D_0)(\pm 10\%) = 1.66\ B_{0c}^{1/2} \text{ or } 1.7\ \bar{B}_0^{1/2}$$

$L_{max,\,n}$ in Table 1 is an extrapolated value of the throw, found to be the best value for correlating the data on centerline velocities with travel.

TABLE 1—EMPIRICAL EQUATIONS WHICH CORRELATE TEST DATA PRESENTED HEREIN ON DOWNWARD JETS OF HEATED AIR FROM A 4-IN. LONG-RADIUS *ASME* NOZZLE[a]

Equations for Isothermal Jet:
For $L/D_0 \geq 8$, $U_c/U_{0c} = 6.4/(L/D_0)$
For $3 < L/D_0 < 8$, $U_c/U_{0c} = -0.008\ (L/D_0)^2 + 0.048\ (L/D_0) + 0.928$
Equations for Heated Jet: n = ½
$L_{max,\,n}/D_0 = \sqrt{2B_{0c}}$
For Primary Zone, $U_c/U_{0c} = \{1 - 2g\,[(T_{0c} - T_a)/T_a]\,[L/U_{0c}^2]\}^{1/2}$
For Principal Zone, $U_c/U_{0c} = (bh_p/h)\ \{1 - [h/h_{max,\,n}]^2\}^{1/2}$
$b = 1/\{1 - [h_p/h_{max,\,n}]^2\}^{1/2}$
$h_p/h_{max,\,n} = 84/(B_{0c} + 138) + 0.06$
$L_p/D_0 = \{172/[1 + 172/(\sqrt{2B_{0c}} + 7.5)^2]\}^{1/2} - 7.5$
$(T_c - T_a)/(T_{0c} - T_a) = h'_p/h' = (L'_p + h'_0)/(L + h'_0)$
$L'_p/D_0 = 1.48\ B_{0c}^{0.14}$
$h'_0/D_0 = 4400/(B_{0c} + 820) - 0.51$

[a] See Nomenclature and Glossary of Terms for definitions.

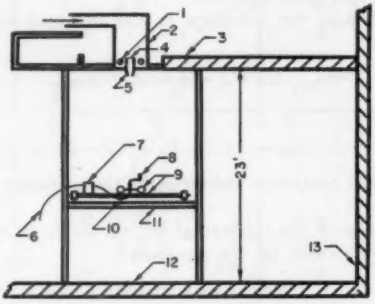
TEST PROCEDURES AND EQUIPMENT

The test rig was similar to that described in Reference 1 (see Fig. 1). The plenum chamber shown in Fig. 1 was attached to the outside of a balcony which extended laterally from the plenum chamber and formed a partial ceiling flush with the bottom of the plenum chamber. Air was delivered through a long duct to the plenum chamber whose base was 23 ft above the floor. A 4-in. *ASME* long-radius nozzle was located at the bottom of the plenum chamber. Screens were installed in the plenum chamber to produce a symmetrical and uniform velocity distribution. In contrast to the previous setups, however, heat was supplied solely by electric light bulbs located around a 1-ft diameter tube, 2 ft long, located inside the plenum chamber immediately ahead of the nozzle. The voltage of the bulbs was controlled with a variable auto transformer. By supplying the heat immediately before the nozzle, the *thermal inertia* of the system was minimized and the temperature of the air leaving the nozzle responded very quickly to changes in the heat supply. This was essential because it proved to be necessary to check and to adjust the temperature at the outlet continuously in order to obtain re-

[2] Exponent numerals refer to References.

producible data. A thermocouple was installed at the outlet to measure the outlet air temperature. Similarly, it proved essential to check and to adjust continuously the plenum chamber pressure, and by this the outlet velocity. During the tests, the heater voltage and fan speed were continuously adjusted to maintain the desired outlet conditions.

The impact tube and the thermocouple for the measurements in the jet were mounted on an electrically driven elevator and carriage as described in References 1 and 2. The impact tube and the thermocouple could be positioned from the floor level with an accuracy of 0.1 in.



1. Electric bulbs.
2. Thermally insulated plenum chamber.
3. Balcony.
4. Screen.
5. 4-in. ASME nozzle.
6. Electric wiring to electronic airspeed indicator.
7. Condenser head.
8. Impact tube.
9. Carriage.
10. Hose connection.
11. Elevator.
12. Floor.
13. Wall.

FIG. 1—TEST SETUP

For measuring and recording the velocity pressures in the jet, an electronic pressure differential recorder (see Reference 3, page 364) was used. A short rubber hose connected an impact tube with the pressure sensitive element of this recorder. This pressure sensitive element consisted of a pressure sensitive capacitance, the so-called condenser head of this equipment. In previous tests, this condenser head was located on a table in the instrument room and was connected to a Pitot tube by means of 2 long hoses, one for the impact pressure, the other for the static pressure. In the present case, the condenser head was located on the elevator at the same level as the impact tube and well outside of the air stream. This arrangement was chosen for the following reasons:

1. The shortened hose connection between the impact tube and condenser head minimized the viscosity *dampening* of the fluctuations and reduced the inertia of the air column.

2. Static pressure measurements, always of questionable accuracy in a turbulent jet, could be dispensed with and a simple impact tube could be used for measuring velocities.

3. The necessity of compensating for variations in the temperature of the air in the hose connection between the impact tube and the condenser head was avoided.

For calibration, the condenser head was connected to an analytical balance modified to measure small pressure differentials. The electronic unit was calibrated twice a day.

TEST RESULTS AND COMPARISON WITH FORMULAS

The tests were run with outlet velocities ranging from 0 to 60 fps and temperatures at the outlet ranging from 0 to 120 deg above the room temperature measured 30 in. above the floor, remote from the jet. The buoyancy numbers (see Nomenclature) ranged from 14.8 to 1925 for heated air. For each set of outlet conditions 2 completely independent tests were run.

For the isothermal jets, the velocities in the principal zone were represented closely by Equation 1.

$$U_c/U_{0c} = 6.4/(L/D_0) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

or, still better, by

$$U_c/U_{0c} = 7.6/(L/D_0 + 1) - 0.034 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

These formulas for the centerline velocities in the principal zone were valid for $L/D_0 \geq 8$.

In the transition zone of the isothermal jet, *i.e.* for L/D_0 from 3 to 8, the data were represented very closely by the parabola:

$$U_c/U_{0c} = -0.008(L/D_0)^2 + 0.048(L/D_0) + 0.928 \quad . \quad . \quad . \quad . \quad . \quad (3)$$

Fig. 2 shows the velocity ratio, U_c/U_{0c}, for heated jets as a function of distance down from the nozzle outlet expressed in terms of outlet diameters. As mentioned before, for each set of outlet conditions 2 completely independent tests were run, the measurements of which are marked x and o, respectively. The solid lines represent values of U_c/U_{0c} as they are predicted by the correlating equations employed. For reference purposes, that part of the jet where the solid line decreases from $U_c/U_{0c} = 1$ to $U_c/U_{0c} = 0$ has been called the principal zone, and the part from the outlet to the beginning of the principal zone has been called the primary zone.

For the principal zone, the correlating equations for U_c/U_{0c}, Equations 4 and 6, were based on, and except for the introduction of the subscript, n, are the equivalents of the equation given in Reference 6:

$$U_c/U_{0c} = [bh_p/h]\,[1 - (h/h_{max,\,n})^2]^n \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

In Equation 4, h_p/h is the usual term for the isothermal jet. The buoyancy force is taken into account by $[1 - (h/h_{max,\,n})^2]^n$, where $h_{max,\,n}$ increases with increasing buoyancy numbers. The factor b is introduced to make $U_c/U_{0c} = 1$ at $h_p/h = 1$, *i.e.* at the beginning of the principal zone.

Writing Equation 4 for $h_p/h = 1$ and $U_c/U_{0c} = 1$ yields:

$$b = 1/[1 - (h_p/h_{max,\,n})^2]^n = 1/[1 - m^2]^n$$

where

$$m = h_p/h_{max,\,n}.$$

Equation 4 can then be written:

$$U_c/U_{0c} = \{h_p/[1 - m^2]^n\}\,\{[1 - (h/h_{max,\,n})^2]^n/h\} \quad . \quad . \quad . \quad . \quad . \quad (5)$$

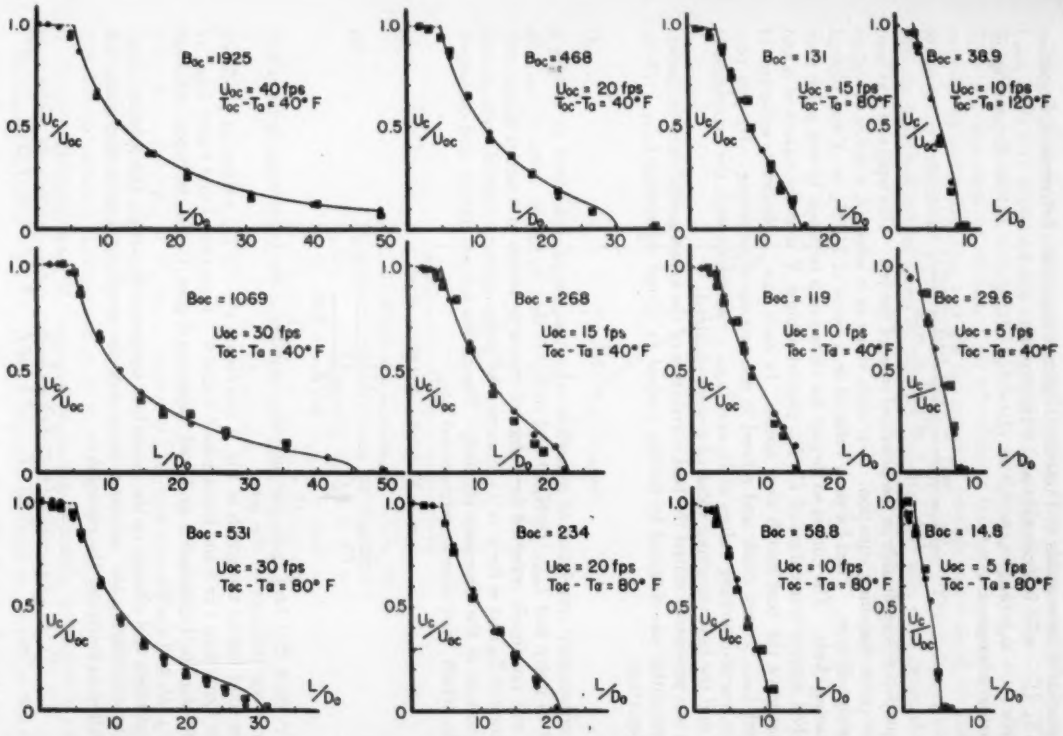

Boc = 1925
Uoc = 40 fps
Toc - Ta = 40° F
Boc = 468
Uoc = 20 fps
Toc - Ta = 40° F
Boc = 131
Uoc = 15 fps
Toc - Ta = 80° F
Boc = 38.9
Uoc = 10 fps
Toc - Ta = 120° F
Boc = 1069
Uoc = 30 fps
Toc - Ta = 40° F
Boc = 268
Uoc = 15 fps
Toc - Ta = 40° F
Boc = 119
Uoc = 10 fps
Toc - Ta = 40° F
Boc = 29.6
Uoc = 5 fps
Toc - Ta = 40° F
Boc = 531
Uoc = 30 fps
Toc - Ta = 80° F
Boc = 234
Uoc = 20 fps
Toc - Ta = 80° F
Boc = 58.8
Uoc = 10 fps
Toc - Ta = 80° F
Boc = 14.8
Uoc = 5 fps
Toc - Ta = 80° F
Uc/Uoc
L/Do

FIG. 2—RATIO OF VELOCITY, U_c, ON AXIS OF JET AT DISTANCE L FT BELOW OUTLET TO VELOCITY AT OUTLET, U_{0c}, PLOTTED AGAINST THE RATIO L/D_0, WHERE D_0 IS THE DIAMETER OF THE OUTLET. SOLID LINE REPRESENTS EQUATION 5, WITH n EQUAL TO ½. THE DOTTED LINE REPRESENTS EQUATION 10

Equation 5 corresponds to Professor Koestel's equation (Reference 4, Equation 3), if $h_p/(1 - m^2)^n$ is a constant and if the exponent n is ⅓.

With $(h - h_p)/(h_{max,\,n} - h_p) = (L - L_p)/(L_{max,\,n} - L_p) = X$ (see Fig. 3) Equation 5 becomes

$$\frac{U_c}{U_{0c}} = \frac{m\{1 - [(1 - m)X + m^2]^n\}}{(1 - m^2)^n[(1 - m)X + m]} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

Equation 6 is applicable in the principal zone of the jet. This equation was used as the curve matching equation. For each value of n selected, a series of values was assigned to m. Then for each value of m, a curve of U_c/U_{0c} *vs.* X was plotted on tracing cloth. The curves so formed for values of n between ⅓ and unity were matched against test values of U_c/U_{0c} plotted against X, and the values of n and m that gave the best match were noted. In each case, preliminary estimates of L_p and $L_{max,\,n}$ were made and refined in the process of matching. By this procedure, curve-matching values of L_p and $L_{max,\,n}$ consistent with the experimental data and the best matching value of n were obtained.

As was previously stated the best correlation of the experimental data on centerline velocities was obtained by setting n equal to ½ and evaluating $L_{max,\,n}/D_0$ by the equation:

$$L_{max,\,n}/D_0 = (2B_{0c})^{1/2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

A satisfactory correlation of centerline velocities was also obtained by setting n equal to unity and L_{max}/D_0 equal to $1.66B_{0c}^{1/2}$, that is, to the value of L_{max}/D_0 that was previously reported for values of throw in excess of 8 outlet diameters[2].

Values of L_p and m for n = ½, obtained by the procedure outlined, were plotted as functions of the buoyancy number. The curves formed narrow and consistent bands, which were closely represented by:

$$m = 84.4/(B_{0c} + 138) + 0.06 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

$$\frac{L_p}{D_0} = \sqrt{\frac{172}{1 + \frac{172}{\sqrt{2B_{0c}} + 7.5^2}}} - 7.5 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

It is evident that here again the buoyancy number is the influencing factor which governs the behavior of the jets.

The solid line in the graph of Fig. 2 represents the U_c/U_{0c} values, as predicted by the Equations 5 or 6 and Equations 7, 8 and 9. Equations 7 and 9 give $L_{max,\,n}/D_0$ and L_p/D_0; Equations 5 or 6 and Equation 8 give U_c/U_{0c} between values of $L_{max,\,n}/D_0$ and L_p/D_0.

Neglecting any change in the centerline temperature within the primary zone, thereby considering the buoyancy force and the resulting deceleration along the centerline as constant, the equation:

$$\frac{U_c}{U_{0c}} = \sqrt{1 - 2g\left(\frac{T_{0c} - T_a}{T_a}\right)\left(\frac{L}{U_{0c}^2}\right)} = \sqrt{\left(1 - \frac{2}{B_{0c}}\right)\frac{L}{D_0}} \quad . \quad . \quad . \quad (10)$$

is obtained for U_c/U_{0c} in the primary zone. The graphs of Fig. 2 show Equation 10 plotted as dotted lines.

Fig. 4 shows the centerline temperature measurements in terms of the temperature ratio, $(T_c - T_a)/(T_{0c} - T_a)$, plotted against distance from the nozzle outlet expressed in outlet diameters.

The vertical dotted line in each graph indicates the point where L/D_0 equals $L_{max,\,n}/D_0$ with n equal to ½. It is obvious from Fig. 4 that $(T_c - T_a)/(T_{0c} - T_a)$ is not yet zero at that point. A clearly significant elevation of temperature, $T_c - T_a$, was observed at $L_{max,\,n}$ for all values of this quantity evaluated with n between ⅛ and unity; *i.e.*, over the full range of possible locations of the bottom of the jet as determined from velocity measurements alone. This indicates clearly that the length of the jet based on velocity measurements was, in all cases, shorter

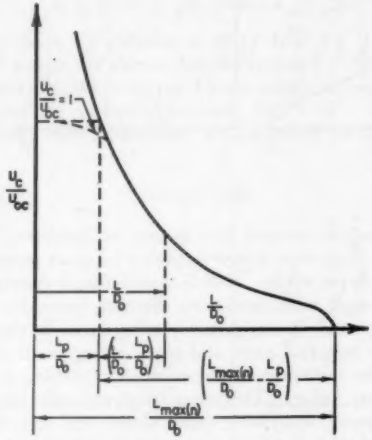


FIG. 3—PLOT OF U_c/U_{0c} VS L/D_0 TO ILLUSTRATE SIGNIFICANCE OF SYMBOLS L_p/D_0, $L_{max,\,n}/D_0$, AND X. $h = L + h_0$; $X = (L - L_p)/(L_{max,\,n} - L_p) = (h - h_p)/(h_{max,\,n} - h_p)$; h_0 = DISTANCE FROM OUTLET TO POINT FROM WHICH h IS MEASURED

than the length based on temperature measurements. For small buoyancy numbers, $(T_c - T_a)/(T_{0c} - T_a)$ had a value of 0.5 or more where L equals $L_{max,\,n}$, n = ½. Between the outlet and this point, the centerline temperature at a given point fluctuated from 2 to 4 deg for the same outlet conditions. Below this point, however, the temperature fluctuated from 15 to 20 deg. Therefore, any measurements in the area below the level of $L_{max,\,n}$, n = ½ can give only approximate values, from which the instantaneous values may differ considerably.

Measured values of $(T_c - T_a)/(T_{0c} - T_a)$ can be fitted to a curve of the form:

$$\frac{T_c - T_a}{T_{0c} - T_a} = \frac{h'_p}{h'} = \frac{L'_p + h'_0}{L + h'_0} \quad \ldots\ldots\ldots \quad (11)$$

To determine the constants L'_p and h'_0, the reciprocal temperature ratio, $(T_{0c} - T_a)/(T_c - T_a)$, was calculated from the measured values and then plotted against

L/D_0. From Equation 11 the reciprocal temperature ratio is:

$$(T_{0c} - T_a)/(T_c - T_a) = h'/h'_p \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

It follows from this equation that a straight line is formed when $(T_{0c} - T_a)/(T_c - T_a)$ is plotted against L/D_0. The intersection of this straight line with the abscissa gives h'_0/D_0. L'_p/D_0 is the value of L/D_0 when $(T_{0c} - T_a)/(T_c - T_a) = 1$.

L'_p and h'_0 determined by this method can be closely represented by:

$$L'_p/D_0 = 1.48\ B_{0c}^{0.14} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

$$h'_0/D_0 = [4400/(B_{0c} + 820)] - 0.51 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

With Equations 11, 13, and 14 it is possible to predict the distribution of $(T_c - T_a)/(T_{0c} - T_a)$. These predicted curves are shown in the graphs of Fig. 4 as solid lines. These formulas should not be applied below the point at which L/D_0 equals $\sqrt{(2B_{0c})}$. In Fig. 4, the curves end at this value of L/D_0. Fig. 4 shows that the equations represent the measured values fairly well.

CONCLUSIONS

The centerline velocities showed fluctuations of 2 different kinds: Fluctuations of the first kind were those with a *high frequency* or short period of the order of one second to a few seconds per cycle. Relative to the local velocity, these fluctuations became bigger and bigger with increasing distance from the nozzle. In absolute magnitude they were small in the primary zone, large in the transition zone and in the first part of the principal zone, and again rather small in the final part of the jet. By recording the instantaneous centerline velocities, it was possible to determine their time mean value. This mean value was called the centerline velocity. Fluctuations of the second kind were much slower and completely irregular. For example, it was found that the time mean values, evaluated over intervals of say 10 sec, might be constant for 2 min, 5 percent lower for the next ½ min, at the original value for the next minute, and again 5 percent lower for several minutes. These irregularities may have been caused by movements of the centerline of the jet. The jet was located in a large hall with tight walls, closed windows, and closed doors and with a relatively uniform temperature in the space surrounding the jet. There was, therefore, no apparent reason for variations in the location of the axis of the jet. Nevertheless, imperceptible variations in the air movement in the hall may have had a considerable influence on the location of the centerline of the jet. The greater the distance down from the nozzle, the greater were the variations in the location of the center of the jet. On the other hand, the velocity profiles became flatter so the variations had less effect on the measured values of velocity. Therefore, the influence of the centerline shifts on the measurements was about the same over all of the principal zone.

Every possible care was taken to make certain that the actual centerline values were measured. Due to the irregular fluctuations, however, no procedure or method for precisely doing this was developed.

The temperature fluctuations influenced the accuracy of the measurements to a much smaller degree than did the velocity fluctuations. The temperature fluctuations were also much more regular with the exception of the fluctuations encountered in that part of the jet below $L_{max,\ n}$, $n = ½$.

Equations 4, 5, and 6 were found to best represent the data when $n = ½$. Actually, the exponent n has a significant influence on the shape of the curve repre-

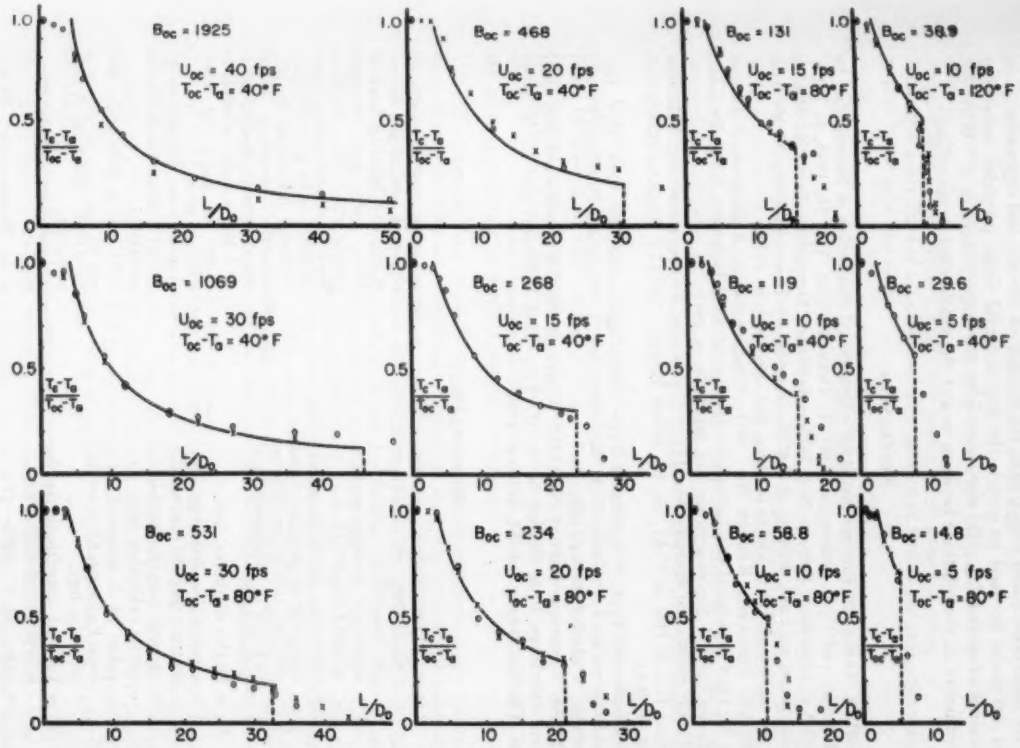


FIG. 4—TEMPERATURE ELEVATION, $(T_c - T_a)$, AT VERTICAL AXIS OF JET EXPRESSED AS A RATIO OF $(T_{oc} - T_a)$, THE TEMPERATURE ELEVATION AT CENTER OF OUTLET, AND PLOTTED VS DISTANCE FROM OUTLET, L/D_0, IN OUTLET DIAMETERS. THE SOLID LINE REPRESENTS EQUATION 11 WITH VALUES OF L'_p/D_0 AND h'/D_0 EVALUATED FROM EQUATIONS 13 AND 14. THE VERTICAL DOTTED LINE MARKS THE POINT AT WHICH $L/D_0 = L_{max,\,a}/D_0$

sented by Equations 5 and 6 only near the bottom of the jet, and exponents of ⅓ or even 1.0 could be used to match the experimental data, provided somewhat larger values of m *and* L_{max}/D_0 were used with the larger values of n. With different values of n, the relations between m and B_{0c} would change accordingly.

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NOMENCLATURE

b = a dimensionless empirical factor = $1/\{1 - (h_p/h_{max,\,n})^2\}^n$.

B = buoyancy number = $(U^2/gD_0)/(T/T_a - 1)$.

c = subscript, denoting values at center of jet, *i.e.*, along vertical axis of jet defined as locus of maximum local velocities.

D_0 = Outlet diameter, feet.

g = acceleration due to gravity, feet per (second) (second).

h = vertical coordinate for locating velocities, feet.

h_p = value of h where U_c/U_{0c} from Equation 5 has a value of unity, feet.

h' = vertical coordinate for locating temperatures, feet.

h'_p = value of h' where $(T_c - T_a)/(T_{0c} - T_a)$ from Equation 11 is unity, feet.

L = distance from outlet, feet.

L'_p = value of L where $(T_c - T_a)/(T_{0c} - T_a)$ from Equation 11 is unity, feet.

m = $h_p/h_{max,\,n}$.

max = subscript, denoting value at observed end of jet, determined by moving both a Pitot tube and thermocouple into the jet from floor level and noting point at which both gave distinct evidence of entering the jet.

max, n = subscript denoting value where U from Equations 4 or 5 has a value of zero.

max, n, n = ½ = same as max, n for case wherein n = ½.

n = exponent in Equations 4, 5, and 6, determined experimentally.

n = used as a subscript, see max, n.

0 = subscript, denoting value at outlet.

T = temperature, degrees Rankine.

T_a = room Temperature, 30 in. above floor at point remote from jet, degrees Rankine.

t = temperature, degrees Fahrenheit.

U = downward velocity, feet per second.

X = $(h - h_p)/(h_{max,\,n} - h_p)$ = $(L - L_p)/(L_{max,\,n} - L_p)$.

GLOSSARY OF TERMS

Principal zone: Zone wherein values of U_c/U_{0c} from Equation 5 decrease from unity to zero.
Primary zone: Zone between outlet and beginning of principal zone.
Transition zone: Zone of actual jet wherein pattern of flow in the primary zone changes to that in the principal zone.

DISCUSSION

H. B. NOTTAGE, Encino, Calif. (WRITTEN): In view of the fact that the maximum downward travels for velocity and temperature, respectively, have been observed not to agree, a clarification of what this may mean is requested in terms of: (1) fundamentals, and (2) application calculations.

Air movements in the surrounding space have been implied to be of considerable importance, at least in the farthest down regions of the jet. What might this mean for application of the data to the heating of occupied spaces where considerable activity exists?

A. KOESTEL, Cleveland, Ohio, (WRITTEN): The value of n = ⅓ recommended in my paper is based on theory which was formulated after experimental observations. Professor Helander's recommendation that n = ½ should be accepted due to his extensive experimental work on downward projected heated jets. The value of n = ⅓ that I used was for the more general case of heated or cooled jets projected upward or downwards. Does Professor Helander care to extend his recommendation that n = ½ to the more general case of vertical non-isothermal jets?

W. O. HUEBNER, New York, N. Y.: I have no specific comment on the paper, which again shows the inherent difficulties encountered in trying correctly to evaluate room air motion, even under simplified conditions.

The research of Kansas State on vertical airstreams has now come to an end, but Professor Helander has promised to review his findings in a final paper. I want to thank him again for his contribution to our knowledge.

One word about the present activities of the Technical Advisory Committee on Air Distribution. We will soon have some definite research proposals for the Committee on Research, on matters where short-term research should give us some much needed answers. We hope that in this research we can continue our pleasant association with our friends at Case Institute and Kansas State, to which we hope to add The University of Buffalo.

AUTHOR'S CLOSURE (Professor Helander): Dr. Knaak has pointed out that when measured velocities on the axis were plotted against travel, L/D_0, and terminal velocities were extrapolated by means of Equation 4, the point of zero velocity so located did not coincide with the point of zero temperature elevation, also obtained by extrapolation. The former, in fact, was always located ahead of the latter. Dr. Nottage has asked what the implications of this may be in respect to (1) fundamentals, and (2) application calculations.

Dr. Knaak used Equation 4 with the exponent n assigned values between ⅓ and unity. He thus obtained a series of values of L/D_0 each of which located the point of extrapolated zero velocity for a particular value of n. These values were denoted as $L_{max,\,n}/D_0$. Two of these values, the first for n assigned a value of ½, the second for n assigned a value of unity, are of particular interest. These values are: (a) $L_{max,\,n}/D_0 = \sqrt{2B_{0c}}$, indicated by the vertical dotted line in Fig. 4, and (b) $L_{max,\,n}/D_0 = 1.66\sqrt{B_{0c}}$, located where L/D_0 has a value 18 percent greater than its value at the vertical dotted line. The latter value of $L_{max,\,n}/D_0$ is the same as the value of L_{max}/D_0.

At L/D_0 in Fig. 4 equal to the first value of $L_{max,\,n}/D_0$: (a) the average temperature elevation was of the order of 40 percent of the outlet temperature elevation (range 20

percent to 70 percent), and (b) there occurred a distinct change in the character of flow in consequence of which the remainder of the jet was characterized by accentuated fluctuations in temperature.

At L/D_0 in Fig. 4 equal to the second value of $L_{max,\,n}/D_0$: (a) the average temperature elevation was of the order of 12 percent of the outlet temperature elevation (range 6 percent to 20 percent); and (b) the temperature gradient was such that the temperature elevation rapidly approached zero.

These results are consistent with our previous findings, which have shown that: (a) the terminal zone of a heated jet projected vertically downward from an *ASME* nozzle is unstable and characterized by marked fluctuations in temperature and a pulsating flow which at L/D_0 equal to $1.66\sqrt{B_{0c}}$, plus or minus 10 percent, produces the smallest detectable mean reading on a pitot tube; and (b) L/D_0 equal to $1.66\sqrt{B_{0c}}$, denoted as L_{max}/D_0, is the mean distance from the outlet to the point on the axis at which a thermocouple moved upward into the stream first presents evidence of a continuously fluctuating temperature and an accentuated temperature gradient.

From the point of view of application calculations, the foregoing suggests that $L/D_0 = \sqrt{2B_{0c}}$ may be regarded as the beginning of the unstable, terminal zone of the jet. The throw equation, $L_{max} = 1.66\sqrt{B_{0c}} = 1.7\sqrt{B_0}$, is based upon an extensive amount of carefully obtained data and is not brought into question by Dr. Knaak's findings. For heated vertical jets from an *ASME* nozzle and throws in excess of 8 outlet diameters, L_{max} evaluated at $1.7\sqrt{B_0}$ locates within a range of plus or minus 10 percent, the point on the axis at which dynamic and thermal effects having application significance terminate.

Where the temperature elevation plotted in Fig. 4 approximated zero, L/D_0, in the main, did not exceed by more than 10 percent the value of L_{max} evaluated as $1.7\sqrt{B_0}$.

The mechanism by which a region of elevated temperature may be sustained below the level at which downward velocities cease to be perceptible and the probable extent of such a region are matters of significant, fundamental interest. Factors involved are the instability of the terminal zone and the puffs of stream air that occasionally are observed to extend below the observed bottom of the jet. For the jets investigated by Dr. Knaak, the region of elevated temperature apparently extended only a short distance below the level where L/D_0 had the value $1.66\sqrt{B_{0c}}$.

We are not prepared to answer Dr. Nottage's question regarding the effect of activity in the occupied space. In order to obtain reproducible and meaningful data, we have found it necessary to avoid taking data when disturbing effects were produced, for example, by someone opening and closing a door. How activity in the occupied space would affect the heat distributing properties of the jet is another matter, and on this we have no data.

We appreciate Professor Koestel's suggestion that, in view of Dr. Knaak's work, a value of n equal to ½ be accepted as the established value. However, we are now engaged in studying the temperature distribution in the entire space affected by the jet and, for that reason, would like to regard the value of n as still somewhat tentative.

We appreciate the comments made by Mr. Huebner. He has been an encouraging leader and it has been a pleasure to work under his direction and that of the T.A.C. on Air Distribution, of which he is the chairman.

On behalf of Dr. Knaak, I thank Dr. Nottage, Professor Koestel, and Mr. Huebner for their contributions to the discussion. I regret that Dr. Knaak could not be here to present his paper and submit a closure.

HONORARY MEMBER—Elected 1957

DOCTOR ARTHUR CUTTS WILLARD

Dr. Arthur Cutts Willard, President Emeritus of the University of Illinois, was born in Washington, D. C., and received his preparatory education in the public schools of the District of Columbia. In 1904, he was graduated from the Massachusetts Institute of Technology and at once began teaching in California. In 1906, he was appointed assistant professor of mechanical engineering at George Washington University. From 1909 to 1913, he was sanitary and heating engineer in the office of the Chief Quartermaster of the Army.

Dr. Willard first became a faculty member at the University of Illinois in 1913 as assistant professor of heating and ventilating. In 1917, he became professor of mechanical engineering and in 1920 was made head of the Department of Heating and Ventilation. He served as acting dean of the College of Engineering from September 1933 to July 1, 1934, at which time he was appointed the seventh president of the University. In 1946 Dr. Willard was made president emeritus of the University.

During the period of his active connection with the University of Illinois, Dr. Willard was always interested in research, especially in the fields of heat transfer, steam flow and warm-air heating. His reputation in heating and ventilating research is world-wide. He was a pioneer in warm-air heating and it is through his untiring efforts that a residence for research purposes was constructed at the University.

Also, throughout his career, Dr. Willard was actively engaged in consulting work in addition to his faculty duties. Well-known among his many contributions in this connection was his work as consultant on ventilation for the Holland Vehicular Tunnel under the Hudson River, which was the first tunnel in the world used for automobiles, and where the carbon monoxide emitted from exhausts posed a difficult problem.

Dr. Willard was also consultant to the U. S. Bureau of Mines, to the Chemical Warfare Service, and to the U. S. Public Health Service in connection with the ventilation of the House and Senate Chambers of the U. S. Capitol. He also was active as a consultant to the Bureau of Subways in Chicago for many years.

Dr. Willard is the author of numerous publications on heating and ventilating, the best known perhaps being the two-volume reference work on the *Mechanical Equipment of Buildings*, in the preparation of which the late L. A. Harding was co-author.

Dr. Willard has been a member of ASHAE for many years, having joined the Society as a member in 1914. In 1928, he was elected president, following service

as 1st vice president in 1927 and 2nd vice president in 1926. Dr. Willard was also a member of the Council of the Society from 1925–29.

He was chairman of the Executive and Publication Committees. From 1927 to 1932 he was a member of the Committee on Research and during the years 1931–37 he served as Technical Adviser for Research. His Society activities also included service on many technical committees including the Guide Publication Committee, Committee on Ventilation Standards and the Committee on Code of Minimum Requirements for the Heating and Ventilating of Buildings.

In 1944, Dr. Willard became a Life Member of the Society. In 1945, he was chairman of the Committee on Research Fund Raising. At the 42nd Annual Meeting in January 1936, Dr. Willard was the recipient of the Society's second F. Paul Anderson Award. Dr. Willard is a member of the Advisory Board of the Society and continues to take an active interest in Society affairs.

In Memoriam 1957

Name	Joined
Edward L. Baker, Salt Lake City, Utah	1945
Albert E. Bastedo, (*Life Member*), Irvington, N. Y.	1919
Alfred C. Buensod, (*Life Member*), New York, N. Y.	1918
William D. Callan, Lockland, Ohio	1944
Charles C. Cheyney, Buffalo, N. Y.	1913
Herbert J. Church, (*Life Member*), Toronto, Ont., Canada	1922
P. Alfred Croney*, Claymont, Del.	1939
Joseph L. Crupper, III, Washington, D. C.	1950
Victor J. Cucci*, (*Life Member*), Brooklyn, N. Y.	1930
Thomas M. Cunningham, Chicago, Ill.	1930
Robert E. Daume*, Pittsburgh, Pa.	1948
Clyde W. Davis*, Huntington, W. Va.	1947
Felix J. di Benga, Montreal, Que., Canada	1951
Richard S. Dill, Washington, D. C.	1939
G. A. Dornheim, (*Life Member*), New York, N. Y.	1906
Robert Lewis Dryer, Seattle, Wash.	1957
James P. Edmondson, San Diego, Calif.	1949
James Emmett, Jr., Skokie, Ill.	1946
Joseph S. Fillo, St. Louis, Mo.	1953
Norman J. Forrester, St. Laurant, Que., Canada	1936
John K. Franklin*, Washington, D. C.	1948
Francis B. Frazee, Austin, Tex.	1946
Eric F. Hyde, Detroit, Mich.	1937
Frank W. Jordan, San Francisco, Calif.	1946
Robert W. Keeton, Chicago, Ill.	1943
William R. Lawson*, Chicago, Ill.	1951
Lee B. Mann, New York, N. Y.	1930
Orville D. Marshall*, Grand Rapids, Mich.	1931

* Died in 1956, notification received 1957.

In Memoriam 1957

(*continued*)

NAME	JOINED
Raymond Martin, Montreal, Que., Canada	1937
A. G. Matousek, Schuyler, Nebr.	1937
Henry C. Meyer, Jr., (*Life Member*), Montclair, N. J.	1898
Jesse E. Miller, Mission, Kans.	1951
George R. Murphy*, Cincinnati, Ohio	1947
Gordon S. McKernan, Peterborough, Ont., Canada	1942
Jamie O. McLean*, Raleigh, N. C.	1955
William McNamara, St. Paul, Minn.	1930
Oliver E. Nesmith*, Bloomington, Ill.	1928
Fred W. Nolte, Fairfield, Calif.	1952
J. F. Oelgoetz*, Columbus, Ohio	1938
Glenn S. Peacock, Pittsburgh, Pa.	1939
Morris Portnoy, New York, N. Y.	1955
A. Gordon Ritchie, Toronto, Ont., Canada	1933
Martin Schaar*, Milwaukee, Wis.	1947
Harold C. Schmidt, Long Island City, N. Y.	1949
Alexander S. Smith, Denver, Colo.	1944
A. Bowman Snavely, Hershey, Pa.	1937
Christian Sorenson, Indianapolis, Ind.	1948
Albert V. Staniforth, Hull, Eng.	1952
Kiser A. Stephenson, Atlanta, Ga.	1941
Robert R. Taliaferro, Sr., (*Life Member*), Lexington, Ky.	1919
Arthur Theobald, San Carlos, Calif.	1937
J. Mortimer Traugott, Philadelphia, Pa.	1930
Horace E. Whitten, (*Life Member*), Boston, Mass.	1924
Dr. C. E. A. Winslow (*Life Member and Presidential Member*), New Haven, Conn.	1932
Richard G. Wolfinger, Harrisburg, Pa.	1952

* Died in 1956, notification received 1957.

Charles-Edward Amory Winslow

1877-1957

Charles-Edward Amory Winslow, 79, pioneering public health expert who was president of the Society in 1945, died on January 9, at the Woodruff Center, New Haven, Conn., following a long illness. Dr. Winslow, who was the first chairman of the Department of Public Health of the Yale University School of Medicine, retired as professor-emeritus in 1945 after 30 years at the university.

The cause of public health was advanced by Dr. Winslow through research, teaching, writing, lecturing and participation in international advisory groups. Even after his retirement, he carried on an active program, including work for the United Nations World Health Organization. He was editor of the *American Journal of Public Health* since 1944, and author or co-author of several volumes, including *Temperature and Human Life*, 1949; *Cost of Sickness and the Price of Health*, 1952; and *Man and Epidemics*, 1953.

Dr. Winslow was born on February 4, 1877 in Boston, Mass., and attended Massachusetts Institute of Technology, where he received his B.S. in 1898 and M.S. the year following. There he studied under the eminent biologist, William T. Sedgwick. His first interest was bacteriology.

His teaching career began in 1900 at M.I.T., where he served as assistant and instructor in sanitary bacteriology, and later as assistant professor of sanitary biology as well as biologist in charge of the Sanitary Research Laboratory. After a one-term interim at the University of Chicago, Dr. Winslow was named an associate professor at the College of the City of New York in 1910. At this time he became curator of public health at New York's American Museum of Natural History, a post which he held along with his other duties for 12 years. Along with D. D. Kimball and others, he was appointed to The New York State Commission on Ventilation in 1913. He also entered government service as director of the Division of Public Health Education of the New York State Department of Health, 1914–15.

It was in 1915 that Yale appointed Dr. Winslow as the Anna M. R. Lauder professor of public health. In later years, he assumed the directorship of the university's John B. Pierce Laboratory of Hygiene, where he studied the physiology of air conditioning. During World War II he engaged in basic research for the Army on the effects of heat and cold.

Dr. Winslow's international reputation was fostered by his work as a member of the American Red Cross Mission to Russia in 1917, and as general medical director of the League of Red Cross Societies at Geneva, 1921. Later he served for 4 years

as an expert assessor for the Public Health Section of the League of Nations. This was a forerunner of his position as consultant for the World Health Organization, which honored him as recipient of the Leon Bernard Medal at Geneva in 1952. Dr. Winslow had helped Professor Bernard, for whom the award is named, to organize the health section of the League of Nations.

Widely honored, Dr. Winslow referred to his receipt of the Society's F. Paul Anderson Medal in 1949 as *one of the great satisfactions of my life.* After he had joined the Society in 1932, he immediately assumed an active role, as member and chairman of numerous technical advisory committees. He became a member of the Committee on Research in 1936, served on its executive committee for 3 years, as its technical adviser in 1939, and vice chairman, 1940–42. His election to the presidency in 1945 was preceded by service as a member of Council and as second and first vice president. He was honored by Life Membership in 1948.

Many technical papers, the product of Dr. Winslow's research on the relation of atmospheric conditions to human health and comfort, appear in the Society's TRANSACTIONS. He was also a frequent contributor to medical and health journals and other scientific publications. Beginning in 1916 he was editor of the *Journal of Bacteriology* until 1944, when he became editor of the *American Journal of Public Health.*

Dr. Winslow, who was named director of the division of public health education of the New York State Department of Health in 1941, was cited by the *American Public Health Association* for his assistance to *the State of New York in the development of what has become an outstanding example of public health administration,* when the *APHA* presented him with its Special Lasker Award in 1952. He had also received the association's Sedgwick Memorial Medal 10 years before, and served as its president.

New York University presented Dr. Winslow with the honorary degree of Doctor of Public Health in 1918, and the University of California selected him as its Rosenberg lecturer, 1941. The Rockefeller Foundation named him as a member of the board of scientific directors of the International Health Division, 1929–30. Dr. Winslow, who was senior sanitarian of the United States Public Health Service (reserve), also served as a member of the Advisory Council of the National Institutes of Health. He was instrumental in founding the Yale School of Nursing and was chairman of the Committee on the Costs of Medical Care.

In addition to being a fellow of the *American Public Health Association* and the *American Association for the Advancement of Science,* Dr. Winslow had served as chairman of the latter association and president of the *National Association of Housing Officials,* the *Society of American Bacteriologists* and *Connecticut Society for Mental Hygiene.* He was a member of the *Society of Experimental Biology and Medicine, American Society of Naturalists, New England Water Works Association,* and Connecticut Academy of Arts and Sciences.

Surviving are his wife, Anne Rogers Winslow, and a daughter, Anne.

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